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DEVELOPMENT AND TEST OF HMPT-500

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General Electric Company
Pittsfield, Massachusetts

December 1974

Technical Report No. 12107

HYDROMECHANICAL POWER TRAINS

FINAL ENGINEERING REPORT

December 1974

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13. ABSTRACT

An HMPT-500 Hydromechanical Power Train was designed for the requirements of a MICV-type vehicle. To satisfy performance and design specifications, an entirely new design concept utilizing only two pairs of hydraulic elements instead of three was developed. This power train differed from previous General Electric units in that reverse and low forward speeds use hydrostatic drive with the more efficient hydromechanical drive being employed for higher forward speed.

The control system for the transmission was based almost entirely on previous transmission controller design concepts and used existing component configurations in a large number of places. Only in those instances where space considerations became overriding were new (but similar) components used.

Two units were fabricated and sent to test. The first unit was given a 400-hour durability test in the dynamometer facility using a Cummins VTA 903 engine rated at 430 horsepower. The second unit was installed in a modified XM-701 vehicle with a 550 horsepower Detroit Diesel 8V71T engine and was operated for 10,000 miles on the General Electric test course. In both test programs, results are highly satisfactory. Efficiency values coincided with predicted values and vehicle operationsl characteristics were outstanding. A number of minor design deficiencies were uncovered and corrected as a result of the durability tests, but no major problem areas were encountered which would prevent the HMPT-500 from achieving all projected reliability goals.

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I. INTRODUCTION

This report documents work performed by the General Electric Company for the U. S. Army Material Command under Contract DAAE07-72-C-0200. The contract required:

1. preparation of a detail design of a hydromechanical power train (HMPT-500) for use in the Mechanized Infantry Combat Vehicle, XM723, having a gross weight of 40,000 pounds.
2. fabrication of a minimum of two HMPT-500 power trains capable of adaptation to any diesel engine in the 400 to 500 horsepower range.
3. performance of the following tests:
   a. a 400-hour dynamometer durability test on one unit, and
   b. a 10,000-mile durability test with a vehicle of approximately 40,000 pounds weight.

The detailed Scope of Work specified in the contract is presented in Appendix A. To meet the stringent size-weight requirements, the HMPT-500 transmission was based upon a new and unique design concept. Details of the concept, design, fabrication, and test are presented in this report.

II. SUMMARY

A hydromechanical power train, designated HMPT-500 (Figure 1), was designed to meet the requirements of a MICV-type vehicle. To satisfy performance specifications while not exceeding package dimensional limitations, an entirely new design concept was utilized. In this design only two pairs of hydraulic elements were required instead of three, but top vehicle speed of approximately 45 mph was maintained. The power train differed from previous General Electric units in that reverse speed and low forward speed used hydrostatic drive, with the more efficient hydromechanical drive being employed for higher forward speed operation.

The control system for the transmission was based almost entirely on previous transmission controller design concepts and used existing component configurations in a large number of places. Only in those instances where space considerations became overriding were new (but similar) components used.

Upon completion of the design phase, two units (plus spare parts) were fabricated and sent to test. The first unit was given a 400-hour durability test in the dynamometer facility using a Cummins VTA 903 engine rated at 450 horsepower as the power source. The second unit was installed in a modified XM-701 vehicle with a 550 gross horsepower Detroit Diesel 8V71T engine and was operated for 10,000 miles on the General Electric test course.

In both test programs, results were highly satisfactory. Efficiency values coincided with predicted values and vehicle operational characteristics were

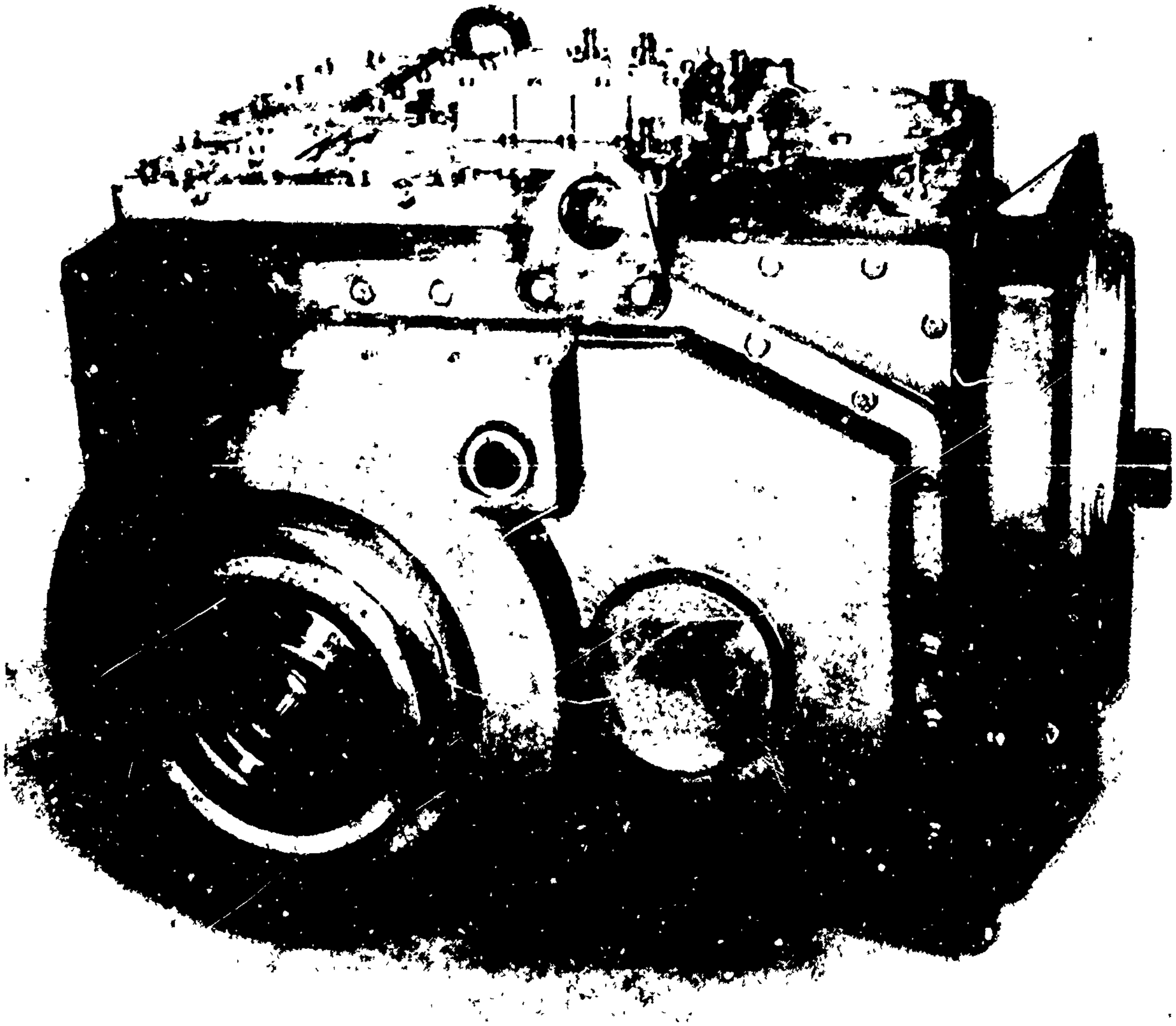


FIGURE 1 Hydromechanical Power Train

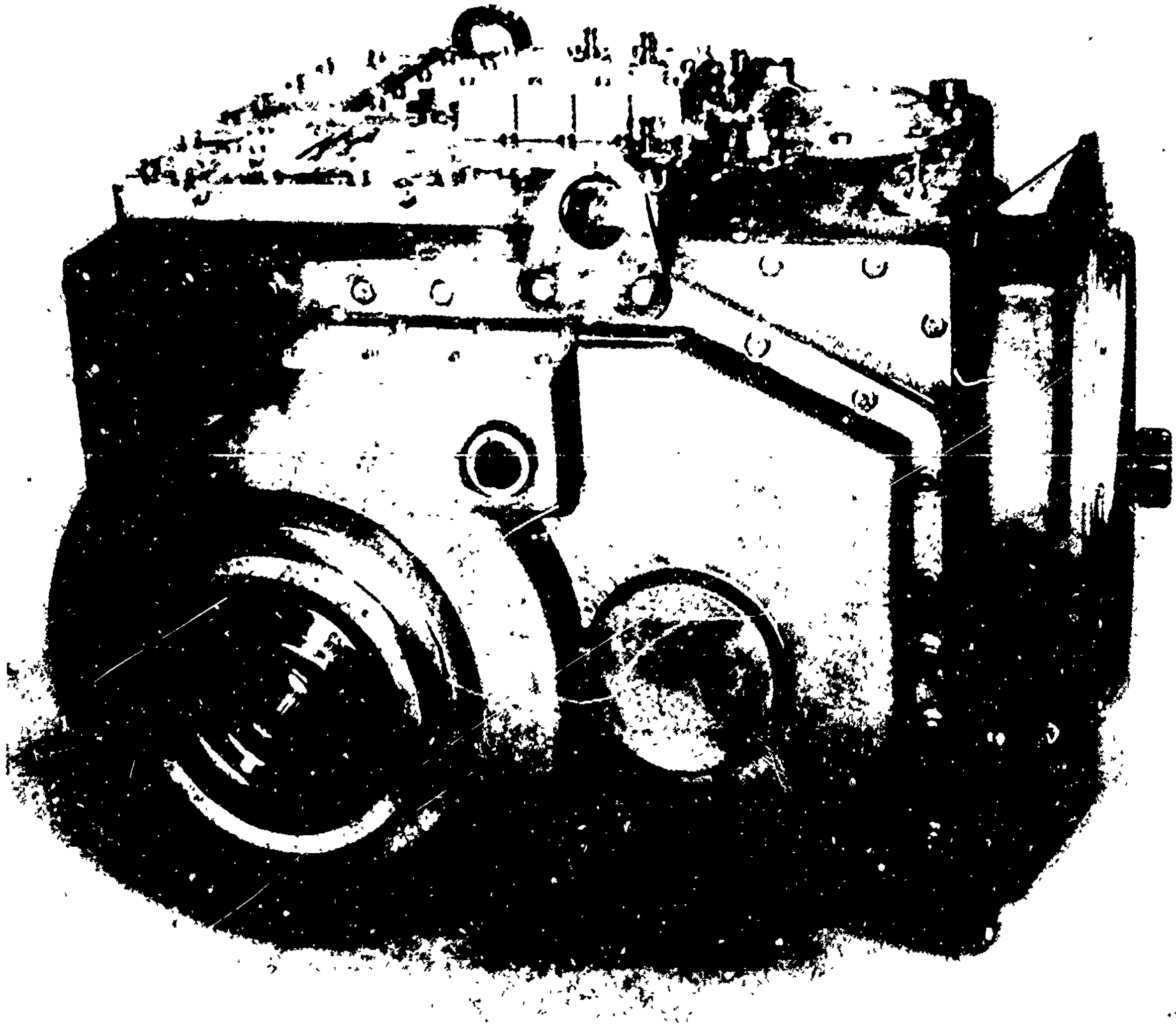


FIGURE 1   Hydromechanical Power Train

outstanding. A number of minor design deficiencies were uncovered and corrected as a result of the durability tests, but no major problem areas were encountered which would prevent the HMPT-500 from achieving all projected reliability goals.

III. TECHNICAL DISCUSSION

A. Design Concept

1. Approach

For many years prior to the design of the HMPT-500, the General Electric Company had studied and refined a building block technique to provide hydromechanical transmission designs capable of meeting any desired torque-speed characteristic. By means of this technique a basic hydraulic element (HMT unit) could be used in combinations of either two or three, with suitable gearing, to give a wide choice of maximum stall torques and maximum output speeds. The highly successful HMPT-100-2 was a typical example of the use of this design concept to produce a 250 horsepower transmission for a 12-ton vehicle.

The HMPT-100-2 employed three identical hydraulic units, the left and right steer units and the overdrive unit. Each hydraulic unit was made up of a variable displacement pump and a motor, both mounted on a common pintle through which efficient close-coupled hydraulic porting was effected between pump and motor. A differential planetary gear set was added to each of the three hydraulic units, making each an efficient differential hydromechanical transmission.

Two of the hydraulic units were placed back-to-back. These were employed for low speed, high torque operation and for steering. To provide for increased speed/torque range, the third hydraulic assembly (overdrive) was added.

For propulsion, the overdrive unit was locked out by the cross-shaft brake during operation under high tractive effort. As higher speeds were called for and the left and right steer units reached their speed capacity, the cross-shaft brake was automatically released and the overdrive unit became operative.

The HMPT-100-2 and all of the other General Electric hydromechanical transmissions were characterized by an infinitely variable propulsion ratio throughout all forward and reverse speeds, infinitely variable fully regenerative steering, reduced system complexity, and excellent overall efficiency.

During 1971 an anticipated need for a hydromechanical transmission to drive a 500 horsepower, 20-ton vehicle at 45 mph or better had prompted design studies to select a suitable transmission using this same building block technique. One disadvantage, already recognized with these designs, was that the use of the third HMT unit to provide the extended top speed contributed to a relatively bulky package. Neglecting all other aspects of weight and cost, the size of the unit seemed to present vehicle installation problems.

Efforts to overcome this drawback resulted in a new and unique transmission concept which used only two hydraulic pump-motor units but provided torques and speeds compatible with projected vehicle requirements. Designated HMPT-500, this new design retained all of the advantages of previous General Electric hydromechanical transmissions with the added advantage of being smaller, lighter, less complex, more efficient, and less expensive than the earlier transmission models.

In 1972 a final design was made to meet the requirements set forth in

Contract DAAE07-72-C-0200 (see Appendix A, Scope of Work). In addition to the specified performance requirements, every effort was made to optimize the design in all areas that would contribute to an outstanding vehicle system. These included:

a. Safety. Personnel and vehicle safety were to be an overriding consideration in all design phases. One area not to be compromised was the previous General Electric Company principle that there be no clutches involved in the steering function so that the transmission will steer without engine power.

b. Vehicle Agility. In addition to the basic infinitely variable steering ratio, both excellent steering response and a large (20 mph) track speed differential were considered to be requirements for the desired vehicle maneuvering capabilities. This has been very effective in the swimming mode of vehicle operations.

c. Ease of Operation. Controls should be designed to minimize driver requirements and training, thereby allowing the operator to concentrate on events outside the vehicle.

d. Fuel Economy. The capability for automatic engine power scheduling should be exploited along with the built-in regenerative steering feature and the high mechanical efficiency to achieve the maximum obtainable fuel economy.

e. Reduced Maintenance. Use of common modules, simple subassemblies, and ease of maintenance were continual goals in making the final design.

f. Durability and Reliability. Proven component designs and a minimum number of parts should receive first consideration in all design areas to enhance an already high inherent reliability.

By combining a new packaging concept using two hydraulic modules (HMT units) instead of three, yet retaining all of the inherent performance capabilities and proven design techniques of previously tested steering transmissions, the HMPT-500 design was undertaken as a major advance in the hydromechanical transmission field.

2. Basic Hydromechanical Configuration

The HMPT-500 is a three-range transmission employing two General Electric ball-piston pump-motor units to provide both propulsion and steer ratios. By a unique gearing arrangement, maximum steering torque is maintained in all speed ranges and, at the same time, the overall ratio coverage is increased in each speed range. As the first step in describing this transmission, the hydromechanical configuration used in each of the three speed ranges will be discussed in the following sections. In all cases, the engine speed is assumed to be constant. Refer to Table 1 for the HMPT-500 Specification Sheet.

a. First and Reverse Range

First and reverse ranges have identical power flow paths, as shown in Figure 2. In this mode, operation is completely hydrostatic. Engine input power is transmitted through the gear train to each of the A-ends or input hydraulic units. A word at this time concerning notation of the hydraulic units would aid in providing an understanding of the pump-motor concept. In this mode the input or A-end is the pump unit and is driven directly from the engine. The output or B-end is the motor and is connected to the sun gear of the output

TABLE 1

HMPT-500 SPECIFICATION DATA SHEET

Input Rating:	Power (net) 500 hp Speed 2600 rpm* (Other input ratios available to match engine speed.) Torque 1100 ft-lbs
Output Rating:	Maximum Torque 9300 ft-lbs Maximum Forward Speed 3260 rpm Maximum Reverse Speed 720 rpm Steering Torque per side 5600 ft-lbs
Weight:	Dry 1630 lbs Oil (not including oil in cooler and lines) 32 lbs
Power Take-offs:	Two available, each with full hp capacity
Brakes:	Service and Parking - mechanically actuated, oil cooled, multiple disk
Dynamic Braking:	Full engine retarding torque plus limited hydrostatic retarding
Steering:	Infinitely variable ratio hydromechanical, fully regenerative, without use of brakes or clutches
Input Disconnect:	Hydraulically actuated clutch to disconnect power train for reduced cold weather engine cranking torque
Control:	Hydraulic control automatically schedules engine speed and transmission ratio for best performance
Hydraulic Fluid:	Mil-L-2104, Grade 30, 0°F to 250°F Mil-O-10295 0°F to -65°F Filtration 40 microns
Hydrostatic Elements:	Four 21 cubic inch/revolution ball piston variable displacement units
Attitude:	70% fore and aft slope 40% side slope
Potential Life:	15,000 miles between overhauls

* 2400 to 3000 rpm can be accommodated in the design

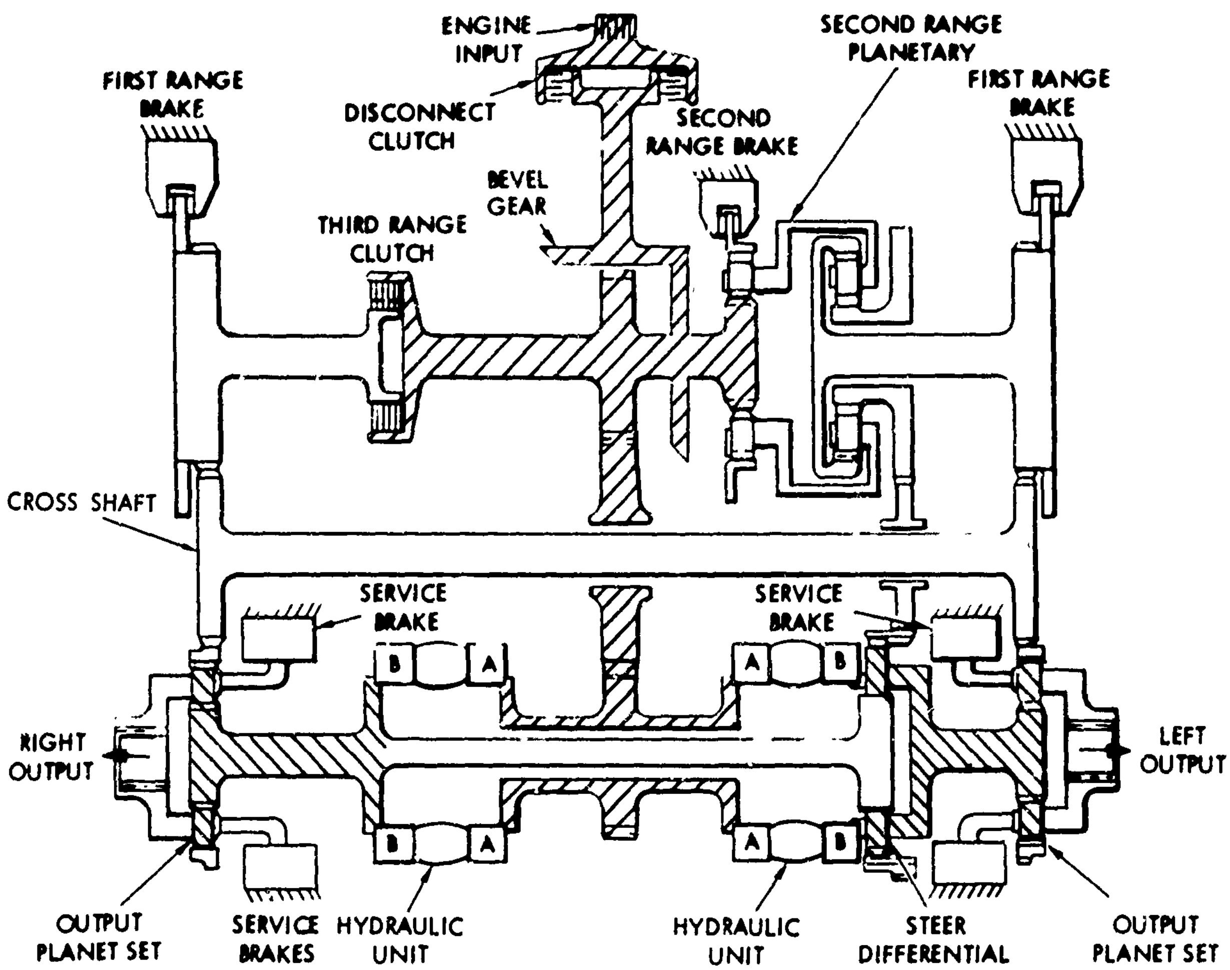


FIGURE 2 Hydromechanical Configuration - First and Reverse Ranges

planetary gear set. In other operational modes this pump-motor function may be reversed. Thus, to avoid confusion, they will be referred to by letter rather than function.

Referring to the right-hand hydraulic unit with the A-end driven by the engine input, the B-end output of the hydraulic unit drives the sun gear of the output planetary set. In first range the first range brakes are applied, thereby locking the cross shaft. The second range brakes and the third range clutch are released. With the cross shaft locked, it can be seen that the ring gear of the output planetary set is also locked. The output planet carrier, which is the transmission output, turns in direct ratio to the hydraulic unit B-end speed.

Operation and output of the left-hand HMT unit are identical. However, the gearing involved merits further explanation since it is one of the primary elements which makes the entire transmission design feasible. The left-hand B-end element is fixed to the left-hand output planetary sun gear and drives it at B-end speed. Physically the connection at the B-end hydraulic unit serves as the planet carrier of a differential gear set so devised that the output speed (ring gear) is always the average of the right-hand B-end output speed (sun gear) and the left-hand B-end output speed (planet carrier). To achieve the desired averaging characteristic, the planet gears are arranged in double-pinion gear sets (Figure 3). Of the meshed double-pinion sets, one pinion meshes with the sun gear and the other with the ring gear. This gearing arrangement serves no

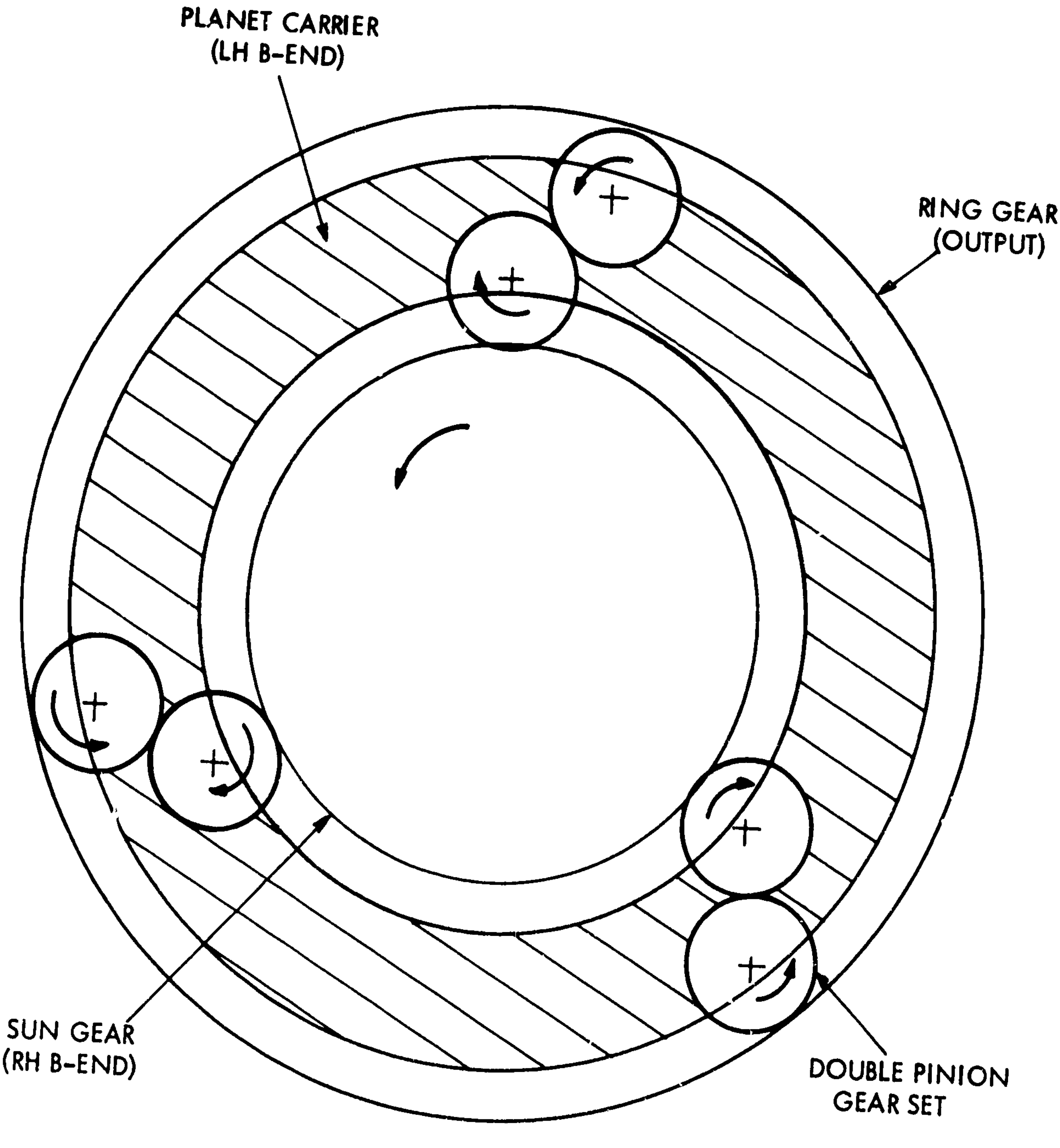


FIGURE 3 Steer Differential Schematic

purpose in first range, but will be discussed later under second range operation.

As described, first range forward and reverse are simple hydrostatic drives with the hydraulic outputs directly connected to the output planetary drives.

b. Second Range

Of the three operating ranges, second range is the most complex to describe. Both second and third range utilize split paths, one hydraulic and one straight mechanical, which combine in the output planetary set to produce the desired output speed. Although it is a temptation to call these power flow paths for descriptive purposes, it leads to confusion since regenerative power loops in both ranges make the true power flow paths under some conditions quite opposite from the expected direction. The term "power flow" path will not be used in any of the following discussions.

In the second range (Figure 4) first range brake and third range clutch are disengaged and second range brakes are engaged. The hydraulic path through the HMT units is identical to first range in that the speeds of the output planetary sun gears are controlled by action of the A-end and B-end hydraulic elements. In the mechanical path two simple planetary sets in series produce an output which turns the cross shaft and this, in turn, through spur gears on each end, rotates both output planetary ring gears. Output speed (output planetary carrier) is a combination of the hydraulically-controlled sun gear and the mechanically-driven ring gear speed.

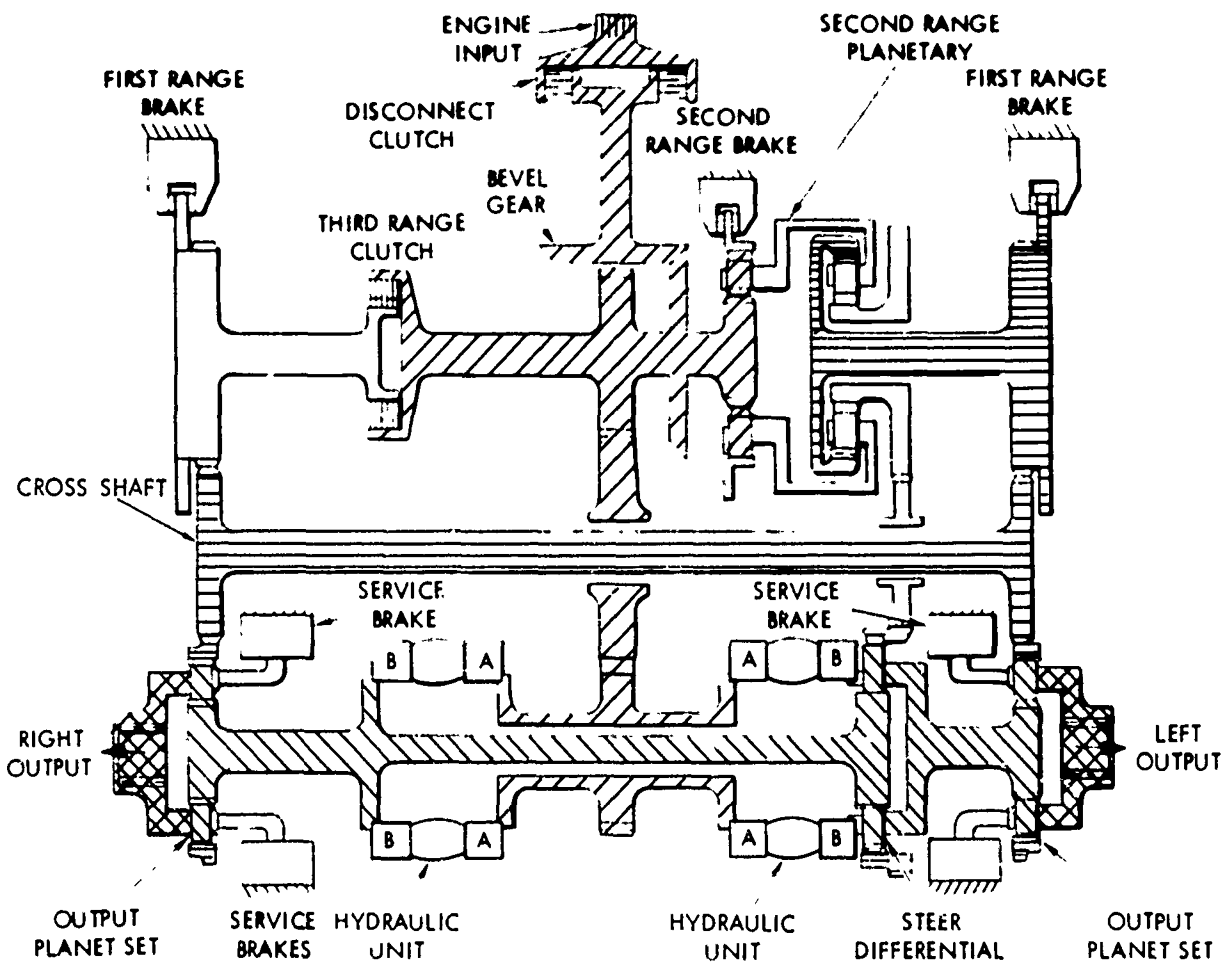


FIGURE 4 Hydromechanical Configuration - Second Range

To return to the two planetary sets in series, the first planetary set has the engine input driving its sun gear, the ring gear is locked by second range brakes, and the planet carrier is the output. This set serves as a speed reducing gear with the added advantage that, by placing second range brakes on the ring gear, the brake torques are minimized.

Output of the first planetary set (planetary carrier) becomes the input to the second planetary set through a common planet carrier for both sets. The sun gear input to the second planet set comes from the steer differential ring gear previously described. This input is the average of the two B-end speeds. No matter what steering maneuver is being made, with one B-end increasing in speed and the other decreasing in speed, the contribution of B-end speed to the second range hydromechanical path is the true average of these speeds. By so doing the infinitely variable ratio in second range remains unaffected by steer maneuvers.

In summary, second range combines a hydraulic and a mechanical path at the output planetary set with the unique characteristic that the input speed from each path is a direct function of B-end speed.

c. Third Range

Third range operation is much easier to visualize. With the third range clutch engaged (Figure 5) and first and second range brakes disengaged, the cross shaft (and the output planetary ring gears) rotates at a constant speed throughout third range operation. All

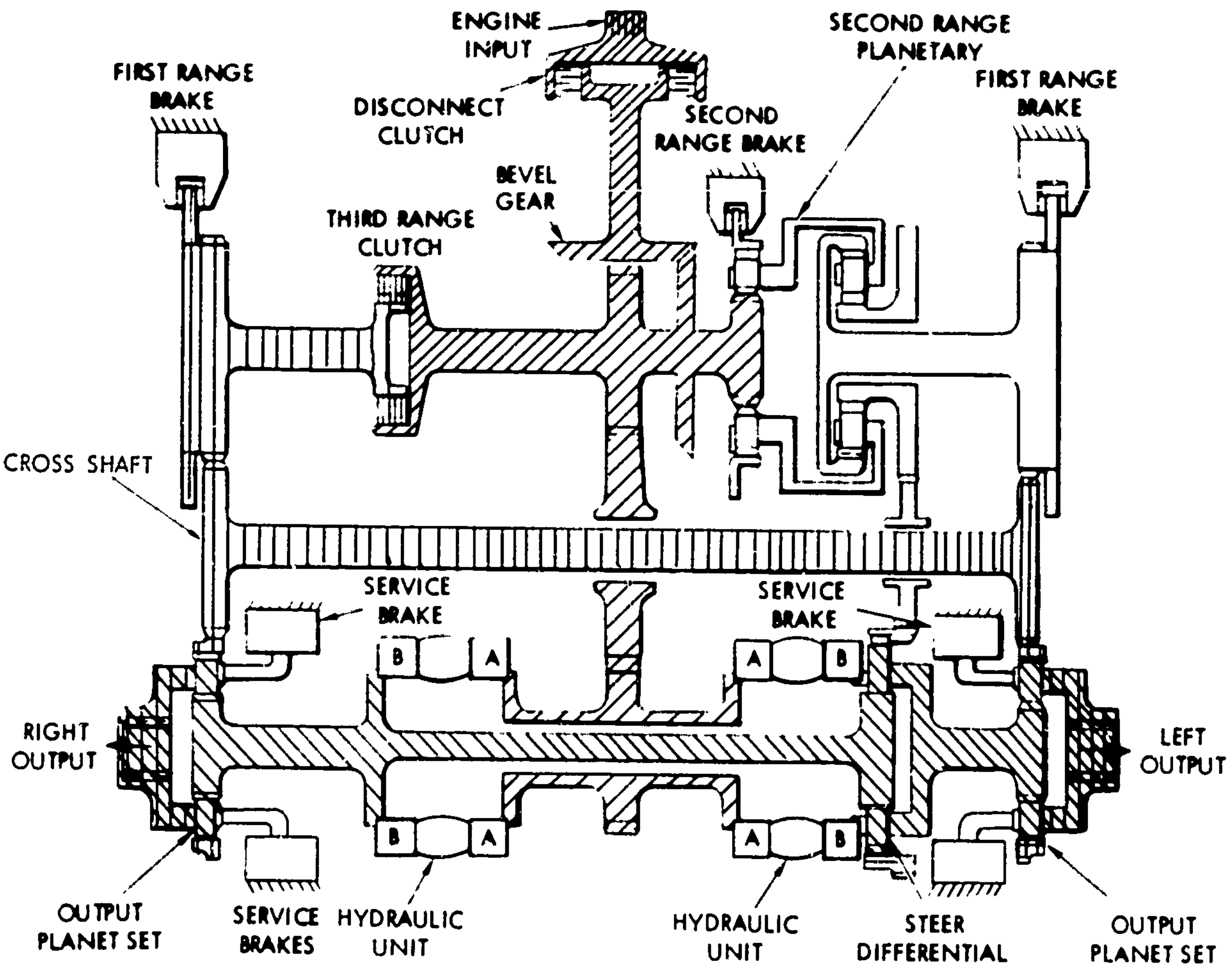


FIGURE 5 Hydromechanical Configuration - Third Range

ratio changes in third range result from variation in output speeds (B-ends) of the hydraulic path.

It should also be noted that steering is identical in all three ranges and meets the design criterion that there be no clutches in the direct path between hydraulic steer command (B-end speed variation) and the transmission output speed. It is also true that the same steer signal results in the same output speed differential in all three ranges. The radius of turn for a vehicle is a function of average track speed divided by differential track speed; for a given steer signal (a track speed differential) the radius of turn increases with increasing vehicle speed. Figure 6 illustrates the minimum radius of turn attainable versus vehicle speed.

3. Mode of Operation

The three basic hydromechanical configurations were described in the preceding section. Operation of the transmission as it passes smoothly through the speed ranges will be discussed in this section and speed characteristic charts for some of the major rotating parts will be shown. A functional description of the controller will be treated separately along with auxiliary functions needed for a complete transmission design.

a. Hydromechanical Section

In each hydraulic unit the direction and magnitude of the B-end output is always dependent on the relative stroking of the A-end and B-end. For the HMPT-500, the B-ends have a fixed displacement while the A-ends may be stroked in either direction from neutral or zero stroke. The actual stroking schedule is shown in Figure 7. Positive

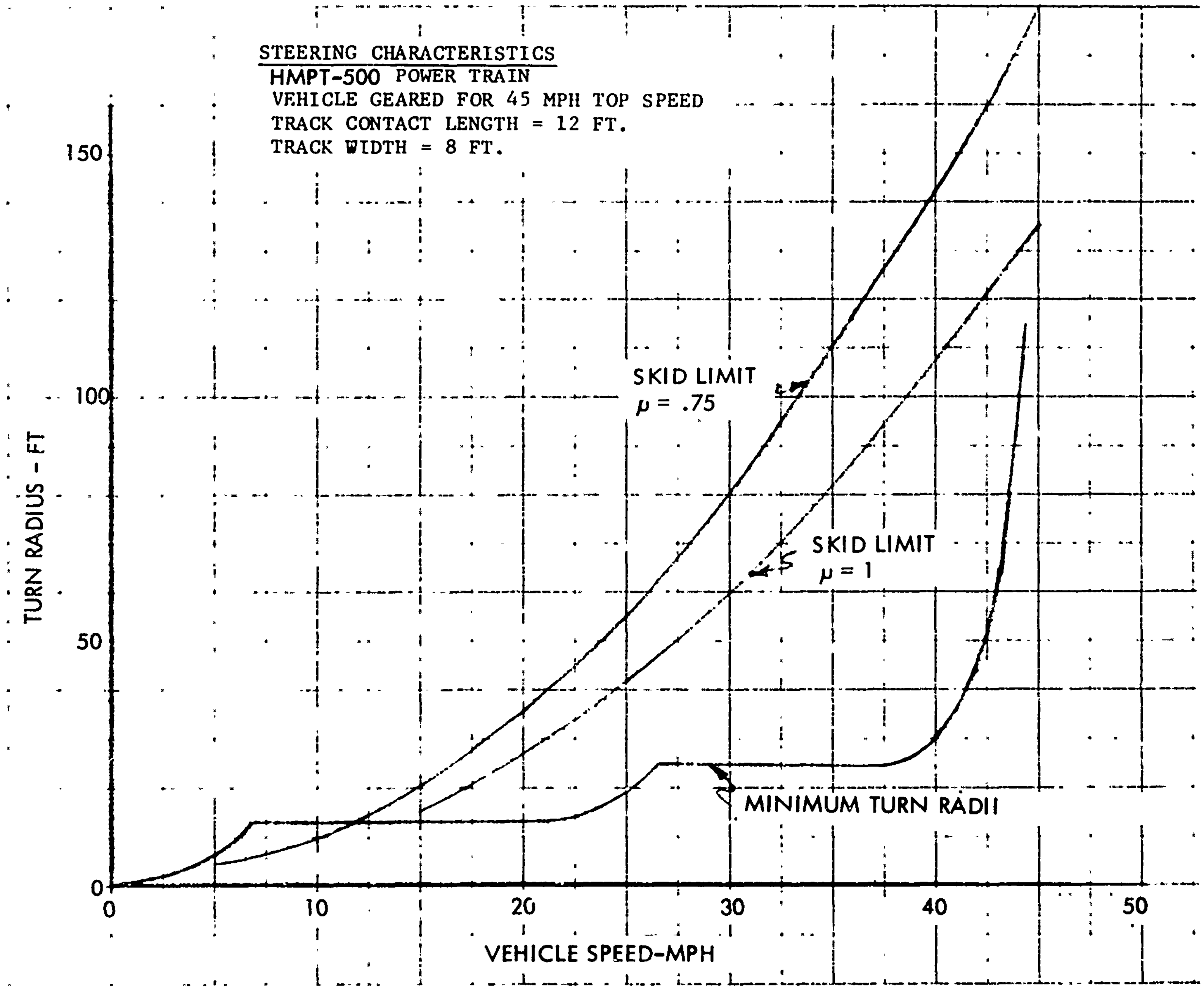


FIGURE 6 Steering Characteristics

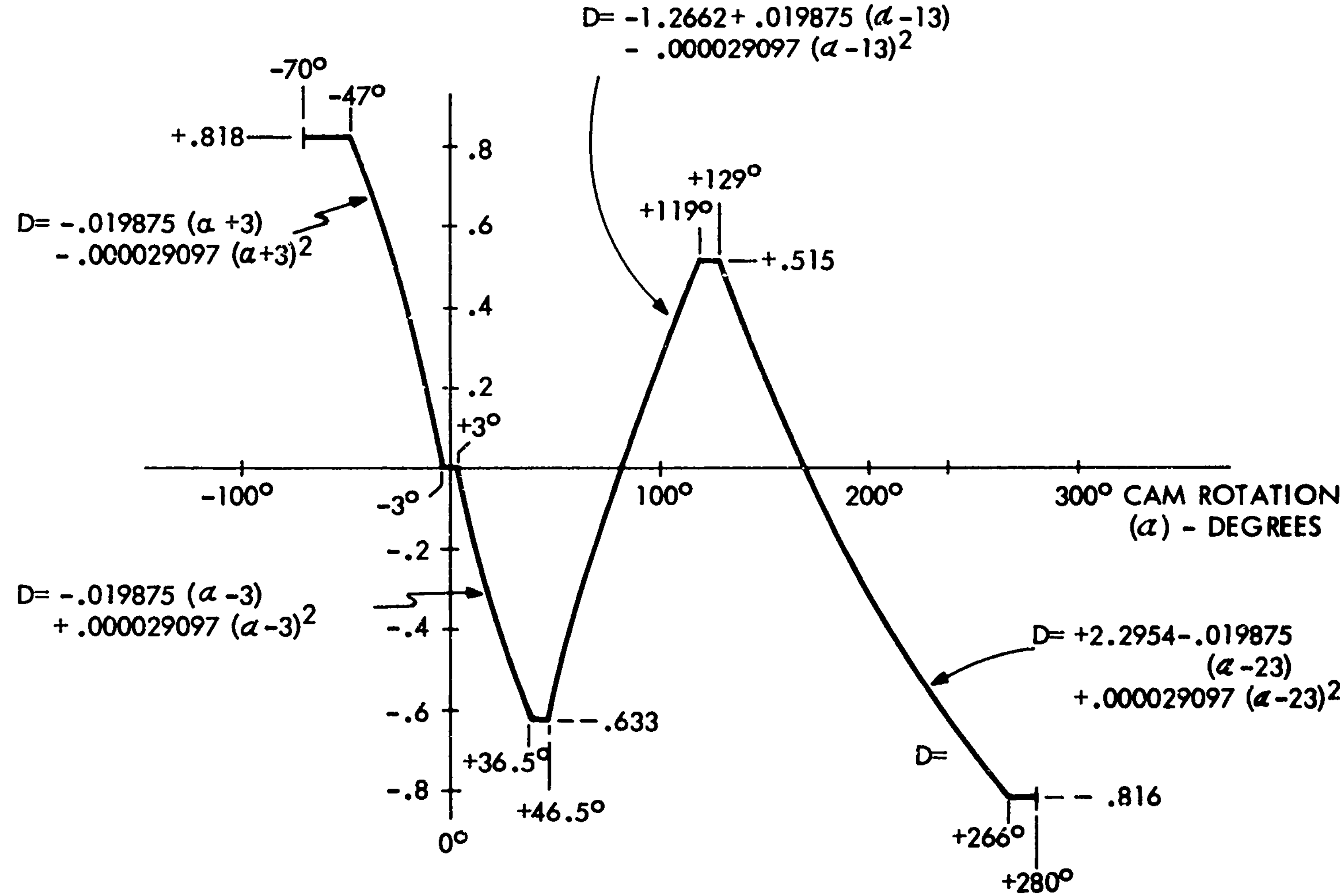


FIGURE 7 Stroking Schedule (cam displacement) for Hydraulic Actuators

and negative sign conventions used in this type of schedule are arbitrary, but plus is taken as the direction the A-end strokes away from neutral as the vehicle moves forward. To tie the stroking schedule to the actual hardware, the direction of positive stroking should also be referenced to some transmission base line. For this particular transmission, positive actuator stroking is toward the transmission input flange.

Operation of the transmission through its complete speed/ratio range is best explained by using a speed characteristic curve (Figure 8) in conjunction with the stroking schedule of Figure 7. On the speed characteristic, four gears are shown with speeds plotted against transmission output speed as the abscissa.

- B-end average - output speed of the hydraulic units which drives the sun gears of output planetary sets.
- Sun #1 - sun gear for first planetary set in second range carrier assembly. Runs at constant ratio to engine speed.
- Ring #1 - ring gear of first planetary set in second range carrier assembly. Carries the second range brake disk.
- Ring #2 - ring gear of second planetary set in second range carrier assembly. Drives cross shaft in second range.

Gear velocities, as plotted on this speed characteristic sheet, represent a constant engine speed of 2600 rpm. Starting at neutral the A-end stroke is zero, B-end speed is zero, and transmission output speed is zero. At the same time, Ring #1 (second range brake) is spinning out at a speed slightly in excess of 2000 rpm, while

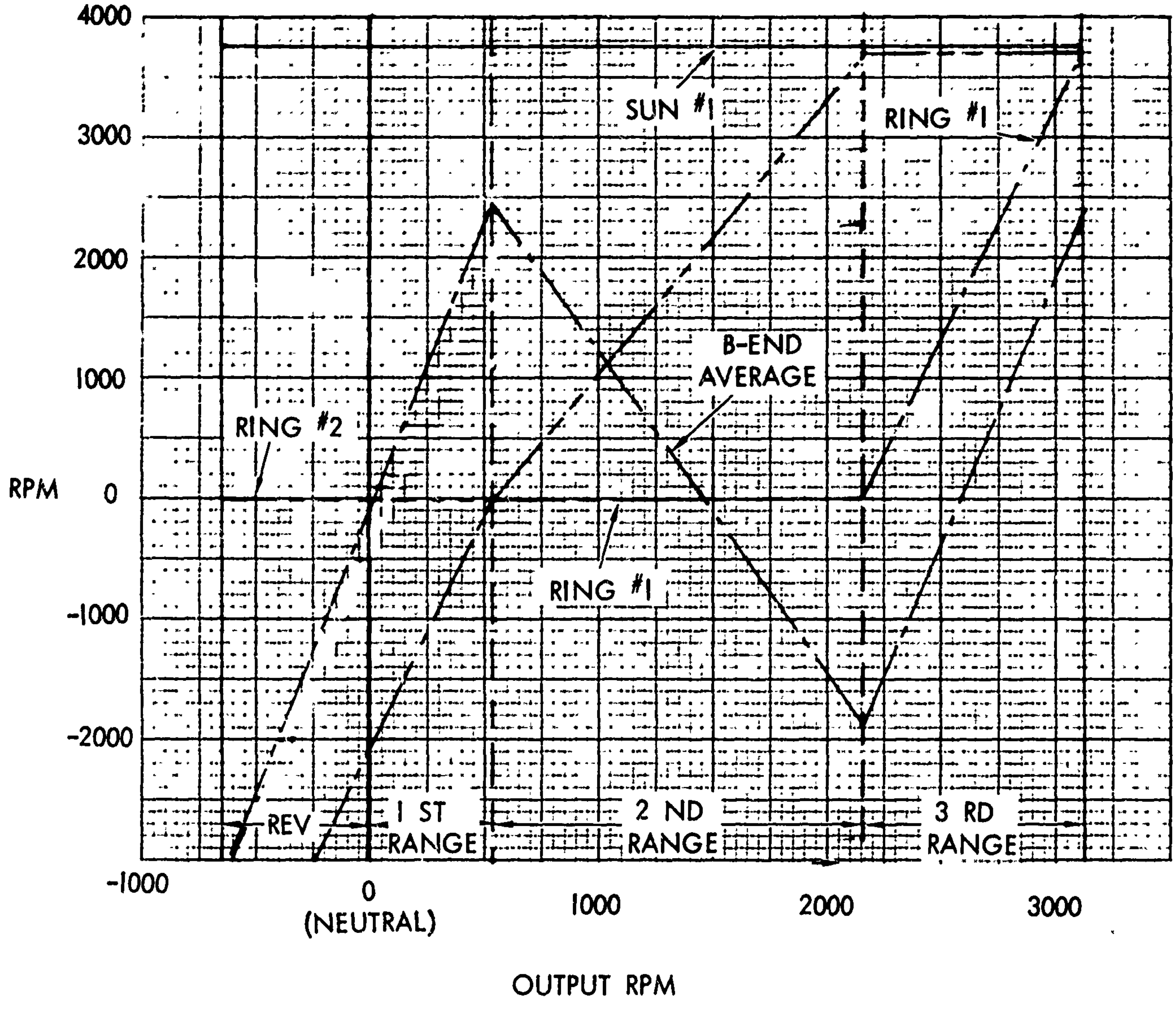


FIGURE 8 Speed Characteristics for Several Typical Gears at 2600 rpm Engine Speed

Ring #2 is locked, preventing the output planetary ring gears from rotating. As the hydraulic units are stroked in a more positive direction, B-end output speed increases, transmission output speed increases in direct proportion, and Ring #1 slows down. This process continues until the actuators have been stroked to plus 80%. At this point B-end speed is about 2400 rpm, output speed about 520 rpm, and Ring #1 speed has become zero. This is the point of a first to second range synchronous shift. Second range brakes are applied (no velocity difference between brake disk and brake pads) and first range brakes are released.

Stroking direction of the actuators is now reversed. The B-end speeds begin to decrease while the now released Ring #2 speed begins to increase. Output speed of the transmission is now the net result of decreasing sun gear speed in the output planetary set and increasing ring gear speed. This results in smoothly increasing transmission output speed with no discontinuicies or ratio gap when passing from first to second range.

The stroking of the A-end actuators continues to decrease, passes through zero, and is stroked in the negative direction. During this period B-end speed decreases, stops, and then begins to turn in the opposite direction. Net output speed of the transmission, however, continues to increase because of the increasing Ring #2 speed. Eventually the Ring #2 speed increases until it is equal to Sun #1 speed. (Note: Sun #1 speed as driven by the engine is constant across the entire ratio range.) At this point relative speed between the two sets of third range clutch plates has reached zero and the synchronous range change from second to third occurs as the third range clutch is applied and the second range brakes are released.

From this point to maximum output speed, the cross shaft and the output planetary ring gears rotate at constant speed. The actuators are stroked back toward zero and increased in a positive direction to a maximum of about 80%. Speed change through the second-to-third range change is again smooth and continuous with no ratio gaps. Output speed is the resultant of the fixed speed ring gear and the changing sun gear (B-end output) speed.

Some increase in transmission output speed could be attained by stroking the A-end hydraulic units to plus 100%. However, by limiting the stroke at the 80% point, added steering capability is maintained at the maximum stroke position which, although unnecessary for maneuvering purposes, provides the driver with a more comfortable feel during high ratio steer conditions.

Reverse operation is identical to first range operation except that the A-end units are stroked to the full minus 100% to achieve the desired reverse speed. No range changes occur in reverse.

By using two hydraulic units and a unique gearing arrangement, a transmission design was achieved that provided smooth and infinitely variable ratio change from full reverse to a maximum forward speed. By means of the three forward ratios, low speed, high torque, and high top speed requirements were met. Steering specifications were met with a system using no clutches in the steer output path, identical steering characteristics in all ranges, and full track speed differential available at every speed from full reverse through neutral and out to maximum top speed. This last characteristic is an important requirement for water operation of the vehicle.

One characteristic of steering which has not been discussed is that of regenerative steering torques. With clutch-brake type steering systems, the energy required to slow down one track is completely dissipated. In General Electric hydromechanical transmissions, the energy generated as the inside track slows down is transmitted directly to the outside or speeding up track. The net result is to minimize power loss with resultant increases in overall operating economy.

b. Controller

The controller design for the HMPT-500 was based on the same basic device successfully used on all General Electric hydromechanical truck and steering transmissions. It is a closed loop servomechanism which maintains engine speed and power to an established reference over a wide range of road loads. The reference is set by the position of the operator's accelerator pedal. Within the controller this pedal signal divides. One signal goes to the controller governor and establishes the speed at which the engine will be governed. The other signal is sent through appropriate linkage to the engine and establishes an engine throttle position or power level.

With this arrangement the engine may be scheduled to follow any power-speed curve that is desirable. Normal practice is to select a curve of minimum specific fuel consumption to achieve maximum economy. Because of the infinitely variable transmission ratio, the selected power schedule can be maintained within very narrow limits.

A block diagram of the HMPT-500 controller is shown in Figure 9. The output of the controller is:

- movement of the two hydraulic unit pilot valves (blocks at upper center of Figure 9) for ratio control and steering.
- operation of three range change pilot valves (left side of Figure 9) in coordination with the stroking schedule.

The function of each of the other blocks will be discussed briefly and related to overall transmission operation.

- Main Governor. Senses the engine speed continuously by hydraulic flow from a speed metering pump and accepts command speed signals from the vehicle operator. It generates hydraulic error signals which are directed to the selector valve and the steer governor circuits.
- Selector Valve. Spool valve is selected by the operator for normal functions (REVERSE, NEUTRAL, and FORWARD) and for emergency functions (PUSH START and TOW).
- Power Piston. Compound hydraulic piston which accepts error signals from the governor and rotates the ratio (stroking) cam through a gear and rack arrangement.
- Ratio Cam and Follower Linkage. Rotating cam with a face cam track for stroking schedule and an edge cam for range change

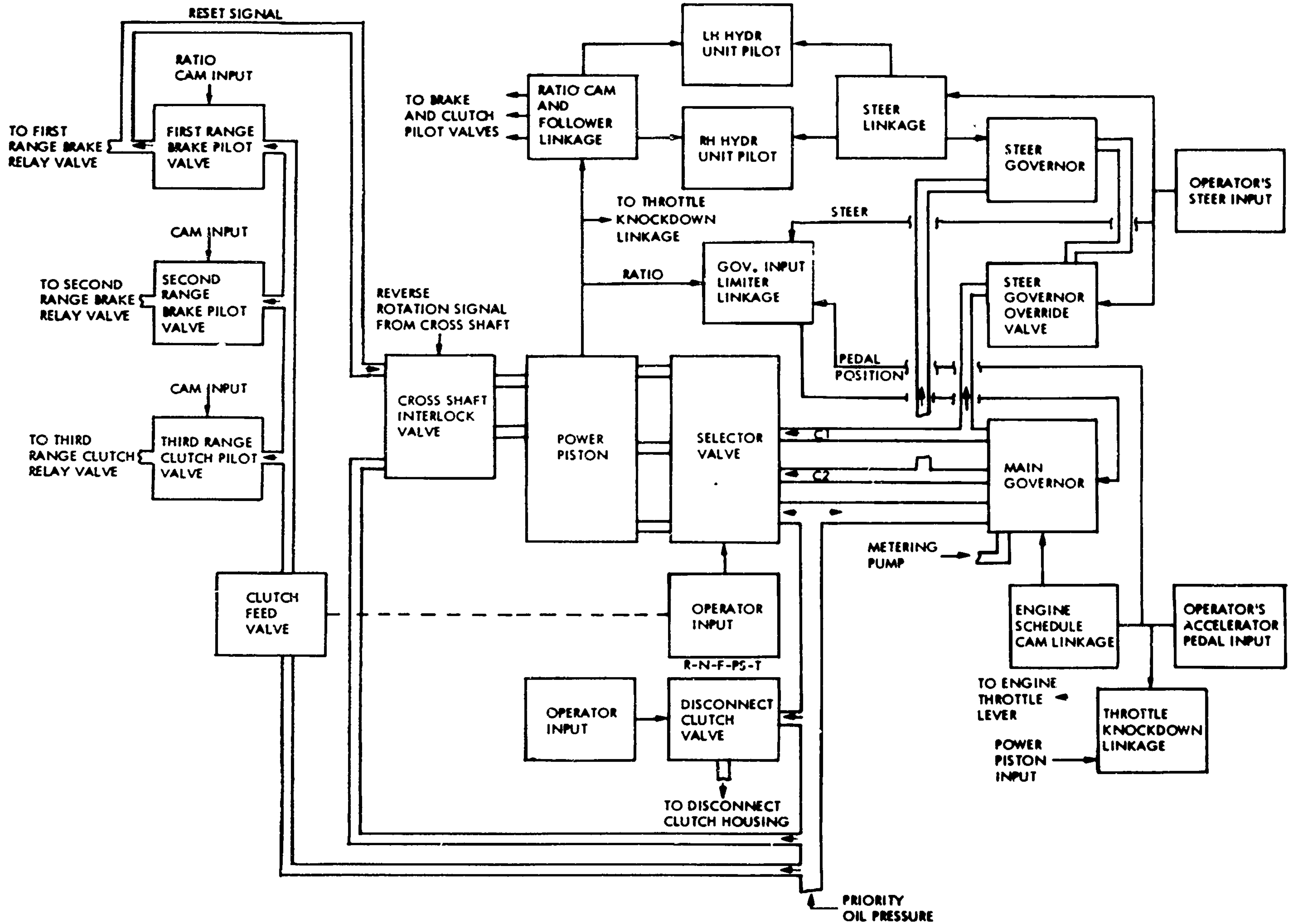


FIGURE 9 HMPT-500 Controller Block Diagram

function. Includes followers which move the hydraulic unit pilot valves and three individual followers for each of the three pilot valves.

- Steer Linkage. Includes variable ratio steer linkage and output steer arm integrally connected to the ratio output linkage.
- Steer Governor. A hydraulic piston accepting main governor error signals. It operates only when stroking ratio is zero (no forward speed) and compensates for steer signals which are exceeding the available power level commanded by the operator's foot signal.
- Steer Governor Valve. Allows a signal to be fed to the steer governor only when steer signal is applied. Provides full steer command when vehicle is stopped in neutral.
- Governor Input Limiters. Signal restrictors for transient throttle conditioning to bias large commands dependent upon transmission ratio and thereby smooth out vehicle operation. One typical requirement is to smooth out acceleration when calling for maximum power and speed from a standing start.
- Engine Schedule Cam Linkage. Cam system in governor input linkage which establishes engine operation along the selected power-speed curve as previously discussed. Accepts different cams for different types and makes of engines.
- Throttle Knockdown Linkage. Allows usage of higher horsepower engines with nominal 500 horsepower transmissions. Limits horsepower from engine at critical points (stall, first-to-second shift, etc.).
- Cross Shaft Interlock Valve. Under high road load conditions in

the low end of second range, hydraulic leakage in the HMT units may cause erroneous singals to the controller. This device senses the condition mechanically and causes the control to make a shift downward into first range.

- Disconnect Clutch Valve. Provides signal to engage and disengage the disconnect clutch located in the transmission input housing.
- Clutch Feed Valve. A valve in the clutch pilot pressure feed line which de-energizes all clutches for TOW operation.

This basic controller design has matured over the years of transmission development and is characterized by its simplicity, lack of complex linkage and hydraulic components, and its adaptability to any of the transmission and engine combinations. As such, its reliability has been demonstrated under a wide variety of field conditions. With this background, it is considered completely capable of meeting all requirements for HMPT-500 transmission.

c. Auxiliary Functions

To this point only the primary propulsion and control functions have been discussed. In any transmission a number of subsystems are required to support these primary functions and to provide other capabilities associated with vehicle operation. Despite the unique features in this transmission, there are no new or unusual requirements for auxiliary subsystems. Designs are, therefore, based on previously developed and proven hardware arrangements. These are discussed briefly and, where applicable, broken down into separate component functions. The hydromechanical schematic (Figure 10) will

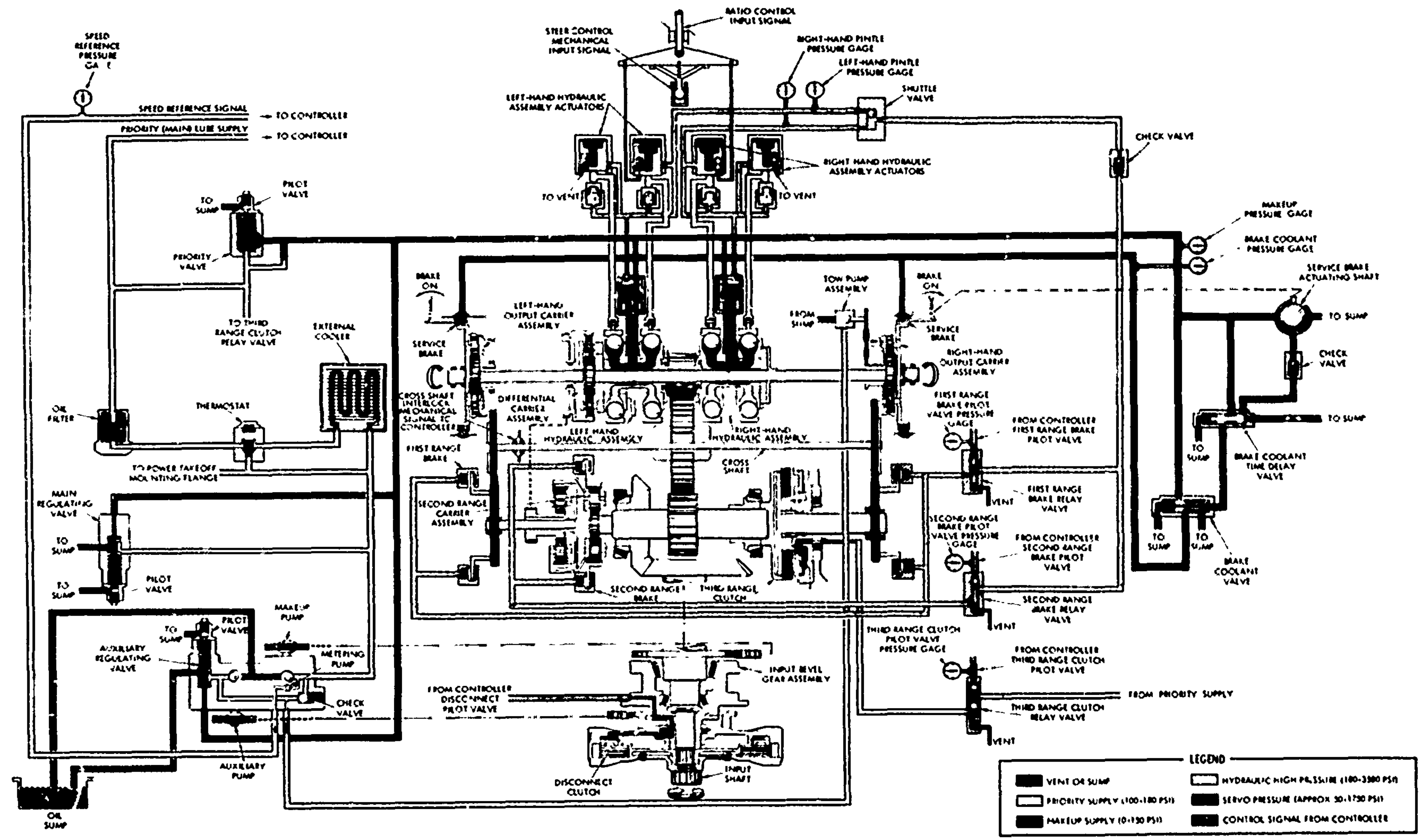


FIGURE 10 Hydromechanical Schematic

be useful for identification of the subsystems.

i. Low Pressure Oil Supply. This is the most complex of the auxiliary subsystems and involves the most functions. It provides low pressure (150 psig nominal) oil, not only to replenish leakage in the main hydraulic unit, but also to supply oil for control functions, service brake cooling, valve and clutch actuation, component lubrication, and the engine speed sensing circuit. Components associated with this system are:

- Auxiliary Make-up Oil Pump - a fixed displacement pump geared to the engine side of the disconnect clutch. This pump provides oil pressure prior to engagement of the clutch.
- Main Make-up Oil Pump - identical to the auxiliary oil pump and operates only when the disconnect clutch is engaged.
- Tow Pump - a smaller, fixed displacement pump used to provide oil for lubrication and control functions when the vehicle is pushed for starting or towed. It reverses automatically for towing in either forward or reverse direction.
- Metering Pump - a small, fixed displacement pump charged directly from the main make-up pump output. It meters oil to the controller at a flow rate directly dependent on engine speed.
- Main Oil Regulator - a piloted pressure regulating valve. It limits pressure in the low pressure oil system to 150 psig.
- Auxiliary Regulator - a piloted pressure regulator identical to the main oil regulator except that it is set to regulate at 120 psig. In conjunction with a check valve, it is used as an

unloading valve for the auxiliary make-up oil pump when oil demand is low.

- Priority Regulator - a third piloted regulator valve installed to ensure that the controller, disconnect clutch, and third range clutch will have adequate oil pressure in preference to all other demands, normal or abnormal, on the low pressure oil system. It is set up at 100 psig and will assure this pressure for the above components at all times, barring actual supply pump failure.
- Thermostat - a typical, thermostatically-controlled oil by-pass valve is used to recirculate cold oil during start-up. This assures rapid warm-up of the transmission. When the oil as reached an operating temperature of about 170°F, the thermostat begins to cause oil circulation through an external cooler.
- Filter - A commercial 40-micron oil filter element is mounted within the main transmission housing. It is located downstream of the external oil cooler and filters all low pressure oil (except metering pump oil) before it is used in the transmission.
- Lubrication System - Although many of the components are lubricated through high pressure oil leakage, oil splash, or drip catchers, many other components (planet gear needle bearings, for example) require a positive lubrication supply. No single network is used for such lubrication. Rather, the make-up oil supply is tapped at convenient adjacent points and controlled by orifices to provide a suitable amount of oil.

ii. Disconnect Clutch. This is a disconnect device between engine and transmission to reduce hydraulic load to a minimum during cold engine cranking. It is a hydraulically-operated, multidisk-type clutch using low pressure oil (priority supply) for actuation.

iii. Clutch Relay Valves. To assure rapid application and release of range change clutches and brakes, the operating valves were placed as close to each of the units as possible. Both first and second range brakes use high pressure (from hydraulic units), while the third range clutch uses low (priority) pressure. All three relay valves are piloted by hydraulic signals from the controller.

iv. Service Brakes. While not a functional requirement from the standpoint of transmission operation, the vehicle service brakes have, historically, been an integral part of General Electric steering transmissions. Brakes are of the multidisk-type and are connected to each transmission output. Operation is mechanical and achieved by rotation of individual actuating shafts on each side of the transmission. Because of the tremendous energy absorption rates required to meet vehicle stopping specifications, the brakes must be cooled. This is accomplished by flowing low pressure oil (make-up supply) through the brakes during application and for about 15 seconds thereafter. Components required to provide cooling are shown schematically in Figure 11 and are:

- Pilot Valve - a rotary valve built into the right-hand actuating shaft. Whenever the brake is applied, a signal from the pilot trips the time delay valve.
- Time Delay Valve - a spool valve having a check valve in the

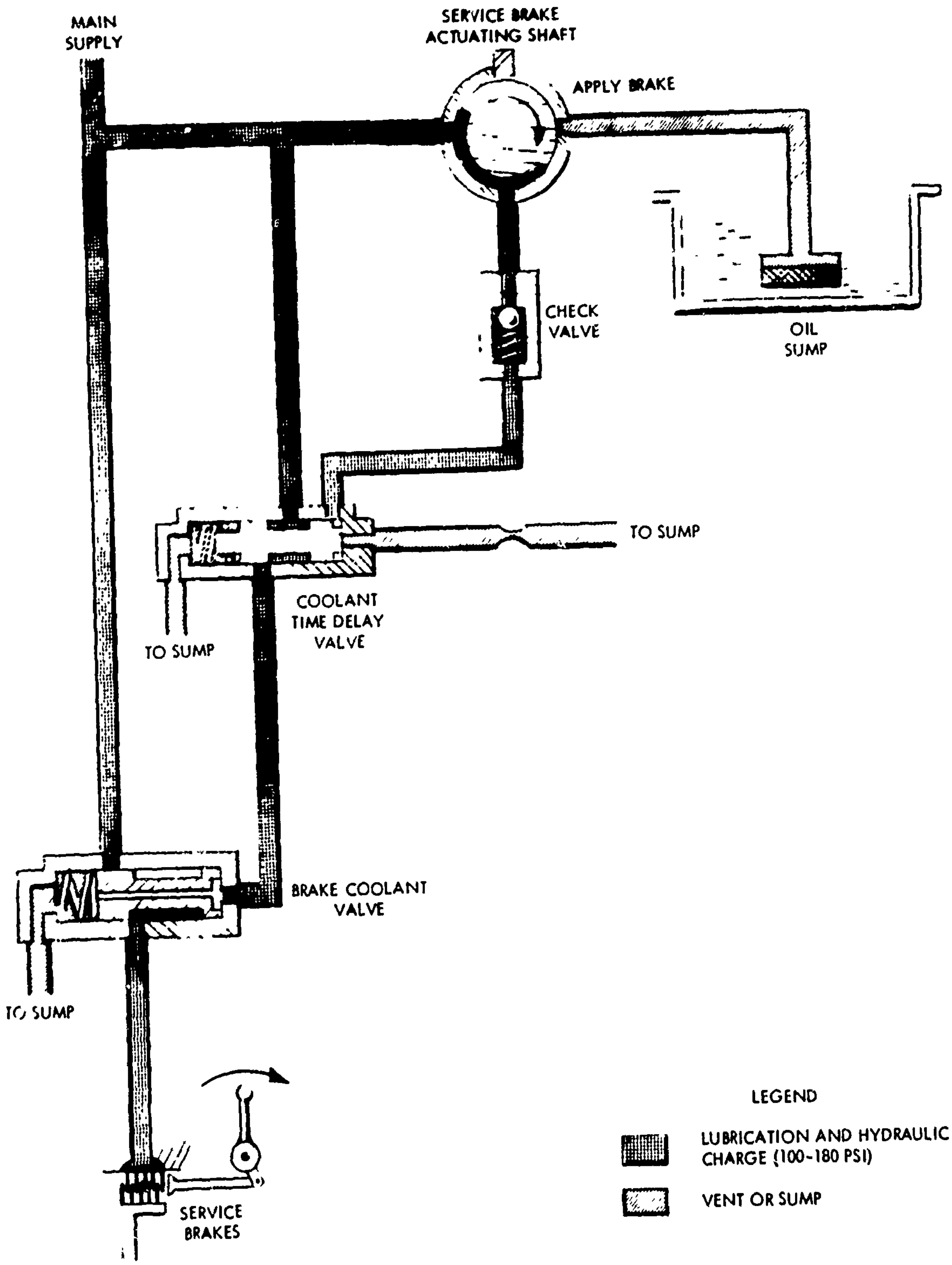


FIGURE 11 Brake Coolant System Schematic

supply line from the pilot valve and an orificed vent. When applied, it transmits a signal to the brake coolant valve. When the pilot signal is removed, the check valve and orifice delay the return of the valve to the OFF position for approximately 15 seconds.

- Brake Coolant Valve - provides large flow of coolant oil to each brake during and subsequent to brake application. To prevent depletion of the make-up oil supply when a high demand is being met elsewhere in the system, the coolant valve is spring-loaded to act as a priority valve. For normal make-up pressures (120-150 psig) coolant will flow unrestricted. If pressure drops below 120, coolant flow is reduced and reaches zero if make-up drops to 100 psig. However, if make-up pressure builds up again within the time delay period, additional coolant will flow automatically.

v. Power Takeoffs. The locations and capabilities of power takeoffs (PTO) are closely related to vehicle designs and requirements and are almost invariably tailored for each specific application. With such definitive information unavailable in early design phases, an arbitrary decision was made to design in two PTO's, one on each side of the transmission. These were to be full power PTO's powered from either end of the engine-driven bevel gear cross shaft.

B. Detail Design

1. System Configuration

The HMPT-500 transmission design utilizes seven major castings. In keeping with the philosophy of simplified maintenance and assembly, each of these castings becomes the basis of a major subassembly. As shown in Figure 12, these consist of the main housing section, left and right intermediate housings, left and right output housings, input housing, and controller. Other than the controller, which can be classed as a single functional subassembly, the subassemblies listed above do not follow functional lines, but are physically selected to make assembly or disassembly of the transmission an easy, straightforward process.

To give a further breakdown in the assembly procedure, the major subassemblies are listed below along with the primary components or subsystems which form a part of each.

a. Main Housing

- Left and right hydraulic units
- Second range brake
- Second range carrier first and second planetary sets (less second planetary sun)
- Input bevel assembly
- Bevel gear cross shaft
- Main cross shaft
- Differential cross shaft
- Steer differential sun and planet carrier
- Low pressure oil system pumps and regulators

b. Right-hand Intermediate Housing

- First range brake disk and gear (right-hand)
- First range brake (right-hand)
- Cross shaft gear

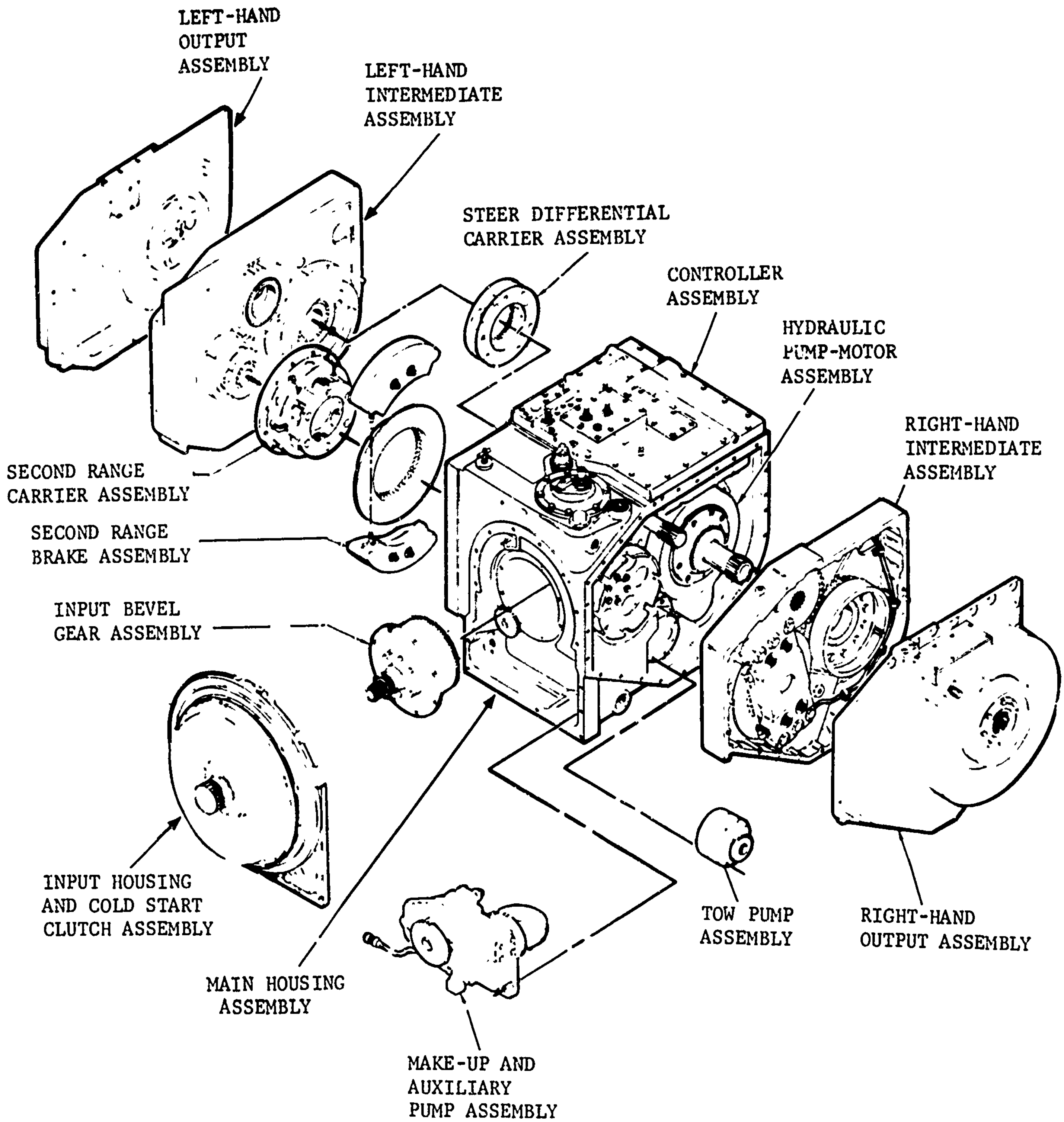


FIGURE 12 HMPT-500 Major Castings (subassemblies)

- Right-hand output ring gear
- Tow pump gears
- Bearing hubs and bearings

c. Right-hand Output Housing

- Right-hand service brake
- Right-hand service brake actuator
- Brake coolant pilot valve
- Tow pump drive gear
- Right-hand output planetary carrier

d. Left-hand Intermediate Housing

- First range brake disk and gear (left-hand)
- First range brake (left-hand)
- Second range second planetary sun gear
- Steer differential ring gear
- Steer differential idler gear
- Cross shaft gear
- Left-hand output ring gear
- Hydraulic lines
- Bearing hubs and bearings

e. Left-hand Output Housing

- Left-hand service brake
- Left-hand service brake actuator
- Left-hand output planetary carrier

f. Input Housing

- Cold start clutch assembly

g. Controller

- All control components

Each of these components is discussed in greater detail in following sections. However, the preceding breakdown is considered important in that it gives a necessary feel for the physical grouping of parts within the transmission and reaffirms a design philosophy aimed at inherently simple assembly and maintenance procedures.

2. Hydraulic (HMT) Units

Two hydraulic units (Figure 13), basically identical except for mounting surfaces, are used in the HMPT-500. Each unit consists of a pintle assembly, two cylinder blocks with ball pistons, two races, an actuator assembly, and several check/shuttle valves for control of both low and high pressure oil.

- Pintle Assembly. The pintle assembly consists of two cylindrical journals on which the A-end and B-end cylinder blocks rotate, the connecting passages that pipe oil between the A-end and B-end units, and the flange section which mounts to the main housing. Because of weight considerations, the HMPT-500 pintles are made from two sections, a nodular cast iron journal section with cast-in flow passages and a forged aluminum flange section, Figure 14. The two sections are shrunk, doweled, and bolted together. The pintle pin on which the two races pivot is also shrunk into the aluminum forging. Both the low pressure check valves and the high pressure shuttle valve use steel inserts to serve as guides. Although in some previous configurations smaller pintles have tended to be made as single, one-piece castings, the idea of two-piece construction is not new and has been extensively tested on one of the earlier General Electric Company hydromechanical transmisions.

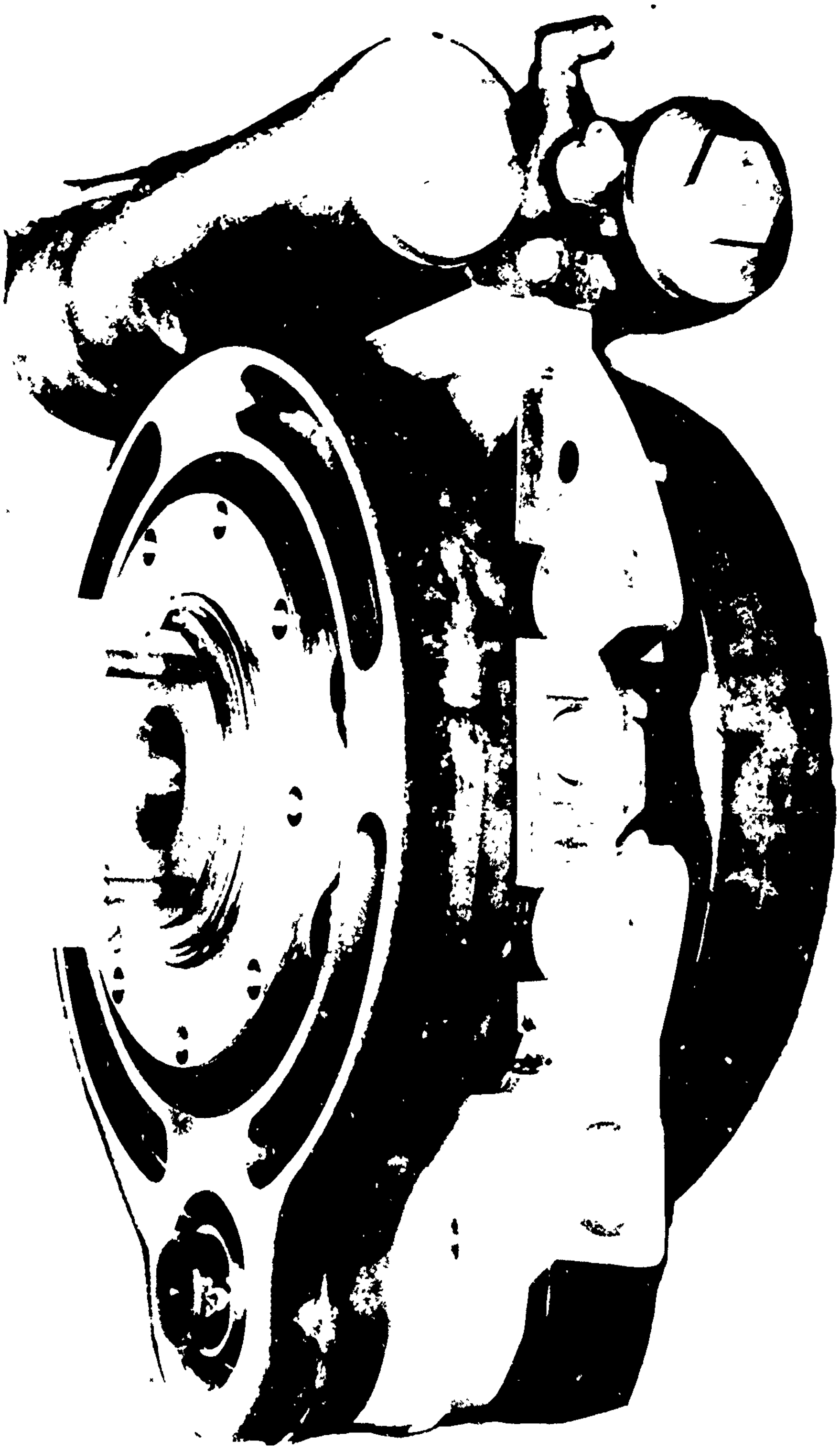


FIGURE 13 Hydraulic Unit Assembly

FIGURE 14 Pintle Assembly

- Cylinder Blocks. The cylinder blocks for both A-end and B-end are nine-bore units using two-inch diameter ball pistons. Each of the nine bores tapers off and passes through the center bushing of the cylinder block. Oil is ported from the pintle flow passages to each cylinder in turn as the block rotates. All cylinder blocks are identical. Flanges to mate with the input gear, the right-hand output shaft, and the steer differential carrier use a bolt and dowel combination to assure alignment and positive retention.
- Races. Each hydraulic unit uses two identical races made from 52100 steel, Figure 15. The A-end unit is strokable, while the B-end is locked at the full stroke position. The races are anchored at the pintle pin and rotate on a spherical bushing pressed into an ear on the race. Opposite the spherical bushing is the race tang on which actuator forces are applied to hold or move (stroke) the race.
- Actuators. The A-end ball piston race is stroked by a hydraulic actuator, Figure 16. On the right-hand side of the race tang is the stroking piston (1) contained within the cylinder housing (2). A pilot valve (3), upon signal from the controller, causes the stroking piston to move either to the right or to the left. The return piston (5), which has half the area of the stroking piston, forces the tang against the stroking piston and balances the forces in the actuator mechanism. Both pistons are spring-loaded to force the race to the center or neutral position when the transmission is not operating.

High pressure oil from the pintle is used to provide pressure for the actuators. Pistons are sized so that the available force always exceeds

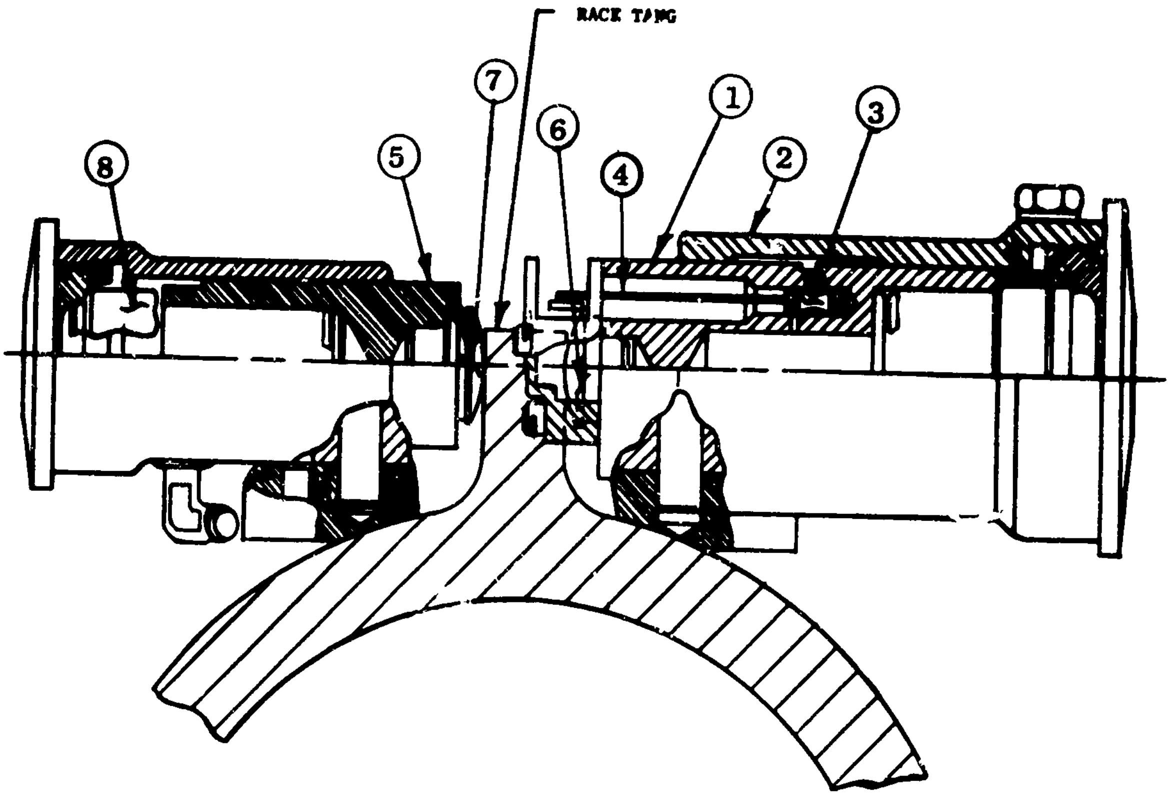
FIGURE 15 Race Assembly

the ball-race forces by an adequate margin. To minimize the contact stresses between the actuator pistons and the race tang, cylindrical contact surfaces are mounted on both pistons.

The B-end race is fixed at a stroke just slightly less than 100% and never varies. Mechanical stops (8) are placed inside the return piston, but hydraulic pressure is also applied to prevent hammering of the tang as forces are reversed within the unit. The large piston is vented at all times. Actuators on both the A-end and B-end are identical and are contained within cylinders that are part of the same casting (Figure 12). Design of this actuator follows closely the design previously developed on the HMPT-100-2 and proven through many thousands of miles of vehicle testing.

- Oil System. Low pressure oil for compensation of ball piston and journal leakage is ported directly to the pintle through the mating flange surface of the main housing. A manifold block and two check valves on the pintle port the fluid to whichever side of the pintle is at low pressure.

High pressure oil is used external to the hydraulic units for actuation of first and second range brakes. A shuttle valve in each pintle (left-hand and right-hand) always selects the high pressure side of the pintle for supplying oil to the actuator. The high pressure from the left-hand hydraulic unit actuator is piped to the right-hand actuator where the two pressures are brought together in another shuttle valve. The higher of these two pressures is ported outside for range clutch actuation.



KEY

1 Stroking Piston
2 Cylinder Housing
3 Pilot Valve
4 Pilot Valve Wire
5 Return Piston
6 Spool Stop
7 Cylindrical Contact
8 Stop

FIGURE 16 Actuator Assembly

- General Configuration. In the transmission assembly the two hydraulic units are arranged such that the two A-end units face each other and are splined to the same engine-driven gears. The B-ends are directly in line with the two output shafts of the transmission. Both sets of actuators are located at the top of the assembly and lie in a horizontal plane. Connection of the hydraulic assemblies to the main housing is accomplished by means of eight one-half inch diameter Grade 8 bolts in each pintle flange with alignment maintained by two dowel pins in each flange.

3. Gears and Splines

All gear design is accomplished through computer programs. These are based upon the classical Lewis formulas modified by stress concentration and material factors (see Appendix B). With these programs, bending stresses are limited to 60,000 lbs per square inch and compressive stresses to 220,000 lbs per square inch. For this transmission the design input torque was 1100 ft-lbs at 2400 rpm (500 horsepower net to the transmission). Torques and speeds for every gear in the transmission can then be computed for use in the appropriate computer program.

To meet durability requirements, all of the gears in the transmission are case carburized steel. To obtain the required case and core hardness, AISI 4620H or AISI 8620 grade of steel is used in all but the bevel gear set. In order to have a high core hardness, the spiral bevel gears are made of AISI4817.

In general, 20° involute spur gears are used in all of the gear trains except the second range and steer differential planetary gears. To maintain proper tooth contact with the 15-tooth planet gears, the pressure angle was increased to 25°.

In order to obtain the highest possible strength, the root fillets of all gear teeth are made with a controlled radius that reduces the stress concentration factor. For quietness and durability the profiles of all gears are controlled in accordance with the best automotive requirements. A finish of 32 is considered acceptable and permits shaving of gear tooth surfaces instead of grinding. The base for all gear tooth geometric requirements is the bearing locating surfaces. By using this base, the gear pitch line run-out is minimized.

All splines are made in accordance with ANS B92.1-1970, Involute Splines and Inspection. Except for the input bevel cross shaft, flat root side fit splines are used. The splines on the input bevel cross shaft are flat root, major diameter fit because of the forces developed in the bevel gear mesh.

Only the splines on the input bevel shaft and the service brake hubs are case carburized. All other splines achieve satisfactory hardness by quenching and tempering to a Rockwell C hardness of 32-45.

The gears, splines, and shafts within the HMPT-500 transmission are representative of conventional practices in the transmission field.

4. <u>Brakes and Clutches</u>

a. Disconnect Clutch

For cold weather starting of a diesel engine, it is essential that the cranking load be reduced to an absolute minimum. To eliminate

- General Configuration. In the transmission assembly the two hydraulic units are arranged such that the two A-end units face each other and are splined to the same engine-driven gears. The B-ends are directly in line with the two output shafts of the transmission. Both sets of actuators are located at the top of the assembly and lie in a horizontal plane. Connection of the hydraulic assemblies to the main housing is accomplished by means of eight one-half inch diameter Grade 8 bolts in each pintle flange with alignment maintained by two dowel pins in each flange.

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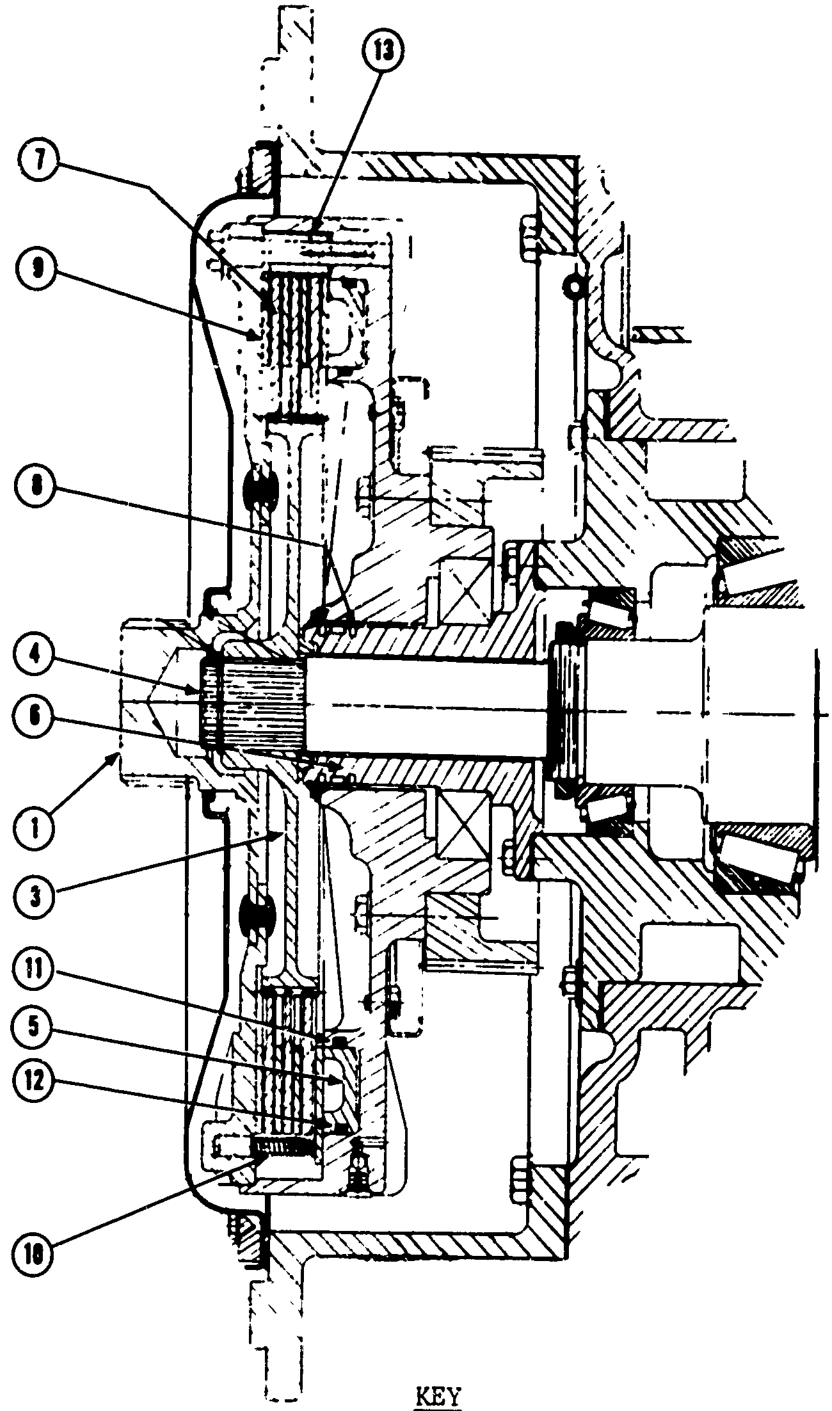
In general, 20° involute spur gears are used in all of the gear trains hydraulic loads induced by cold transmission oil, a disconnect clutch is provided at the input of the transmission. This clutch is a hydraulically-operated, multiple disk clutch based upon a similar design in the HMPT-100-2 transmission.

Referring to the cross section in Figure 17, the input spline (1) is driven by a damper plate connected to the engine flywheel. This drives the main clutch housing (2) which contains four steel pressure plates (9) locked to the housing by detents in the periphery of the plates and bolts which fasten the two sections of the housing together.

The driven side of the clutch has three friction plates (7), faced with Raybestos-Manhattan, Inc. 3672-3 material, splined to a hub (3) which in turn mates with the input bevel gear shaft (4). A ring-shaped piston (5), actuated by priority pressure, forces the clutch plates together for clutch engagement. Oil for clutch actuation is received from a valve in the controller and is piped through the fixed hub (6), which contains two metal sealing rings, to the actuating piston. To assure proper clutch release a centrifugally operated check ball (Figure 18) is used to bleed oil from the clutch piston cavity when the actuating supply (from the controller) is vented. A gear, mounted on the back of the clutch housing, operates the auxiliary make-up pump whenever the engine is rotating and provides a source of pressurized oil for clutch engagement.

In actual practice the clutch is disengaged for all starts, cold or



KEY

1 Input Spline
2 Main Clutch Housing
3 Hub
4 Input Bevel Gear Shaft
5 Piston
6 Hub
7 Friction Plates
8 Hook Ring Seals
9 Steel Plates
10 Spring
11 Seal
12 Seal
13 Spacer

FIGURE 17 Disconnect Clutch Assembly

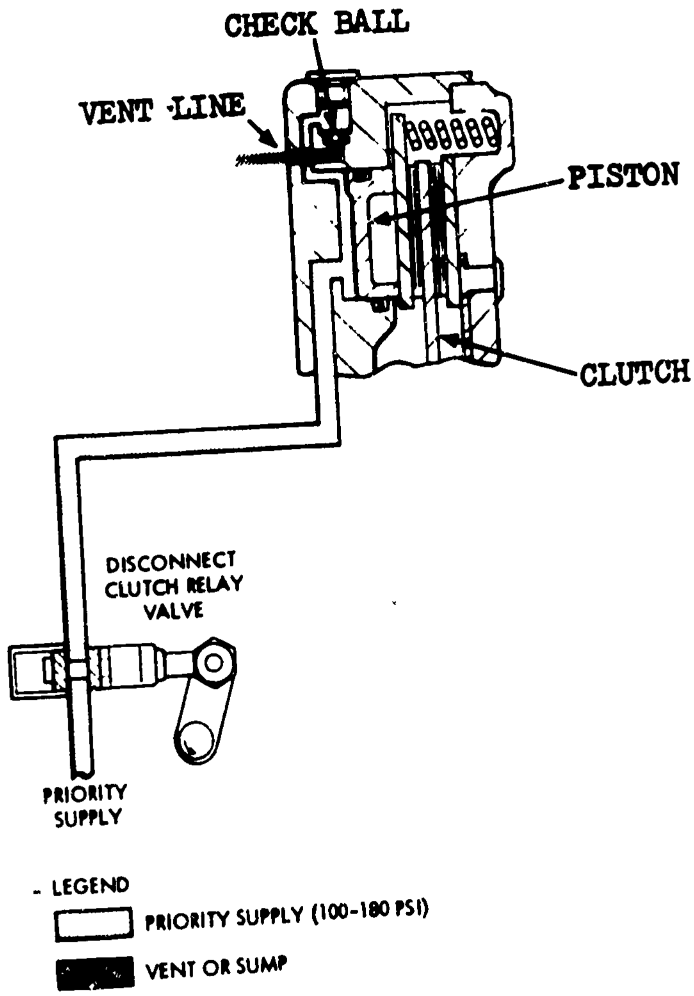


FIGURE 18 Disconnect Clutch Hydraulic System

hot. By doing so, operational safety is increased since neither operator error nor system malfunction can cause unexpected steering or lurching of the vehicle. In addition, the disconnect clutch provides a means for trouble-shooting power plant problems without danger of vehicle movement.

b. First Range Brake

The function of the first range brake is to hold the output planetary ring gears stationary against reaction torque when the transmission is in low range (forward, neutral, and reverse). Two sets of brakes are actually used. These are located in both intermediate housings on the axis of the bevel gear cross shaft. Braking action of each unit locks a spur gear (first range brake gear). These spur gears mesh with corresponding gears on each end of the main cross shaft, while the cross shaft gears in turn mesh with external gears on the output planetary ring gears. When the first range brake is applied, both the cross shaft and the output ring gears are prevented from turning.

First range brakes are of the caliper type, Figure 19. The brake disks are splined internally to mesh with the first range brake gears and can float on the gear as the brakes are applied and released. Four calipers are used on each brake. The fixed disks are mounted in a single forged aluminum housing which straddles the brake disk and is bolted to the intermediate housing. Brake disk material is Raybestos-Manhattan, Inc. material, Type R-1698-7.

The pistons are actuated by high pressure oil from the pintle and are

FIGURE 19   First Range Brake Assembly

released by spring force. A pilot signal from the control actuates a relay valve (Figure 20), centrally located in the main housing, which ports high pressure to both brakes simultaneously. The relay valve is a simple spool valve operated by the low pressure pilot signal from the control and released by a return spring when the pilot pressure is vented. Material of both housing and spool is aluminum. The spool is hardcoated for maximum abrasion resistance. Brake design, except for increased capacity, is very similar to the low range brake system used in the HMPT-100-2 transmission.

c. Second Range Brake

The function of the second range brake is to hold the ring gear of the second range, first planetary set stationary during second range operation. The design is basically identical to the individual first range caliper brakes except for minor hardware details. Instead of floating on a gear as in first range, the second range disk is an integral part of the ring gear which it locks. It was anticipated that any axial motion required could be obtained from movement of the entire ring gear unit.

The fixed disks are mounted in the main housing while the actuated pads are mounted in pairs in two forged aluminum housings which are bolted to the main housing, Figure 21. High pressure oil is again used to actuate the four brake pistons which are spring-loaded for release. The relay valve which ports high pressure to the pistons is identical to that used in first range. Brake disk material is the same as first range.

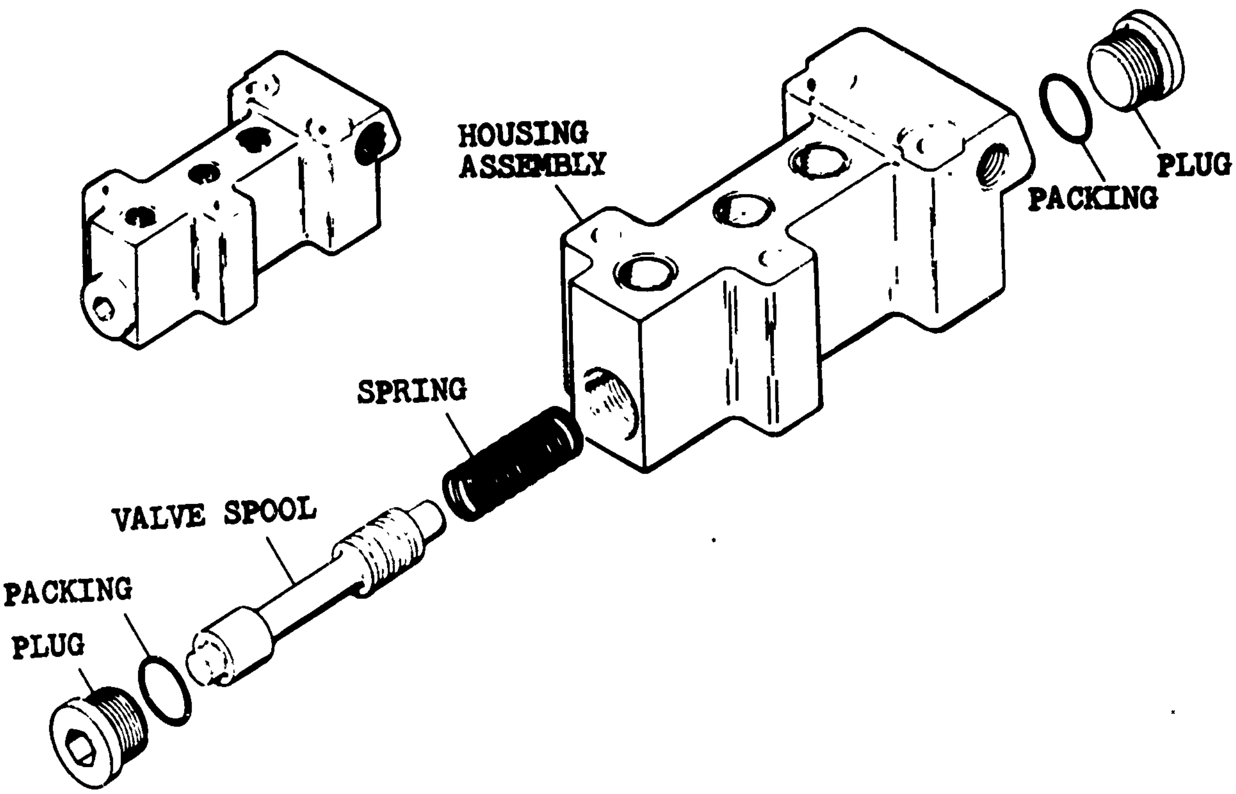


FIGURE 20 First and Second Range Brake Relay Valve Assembly (exploded view)

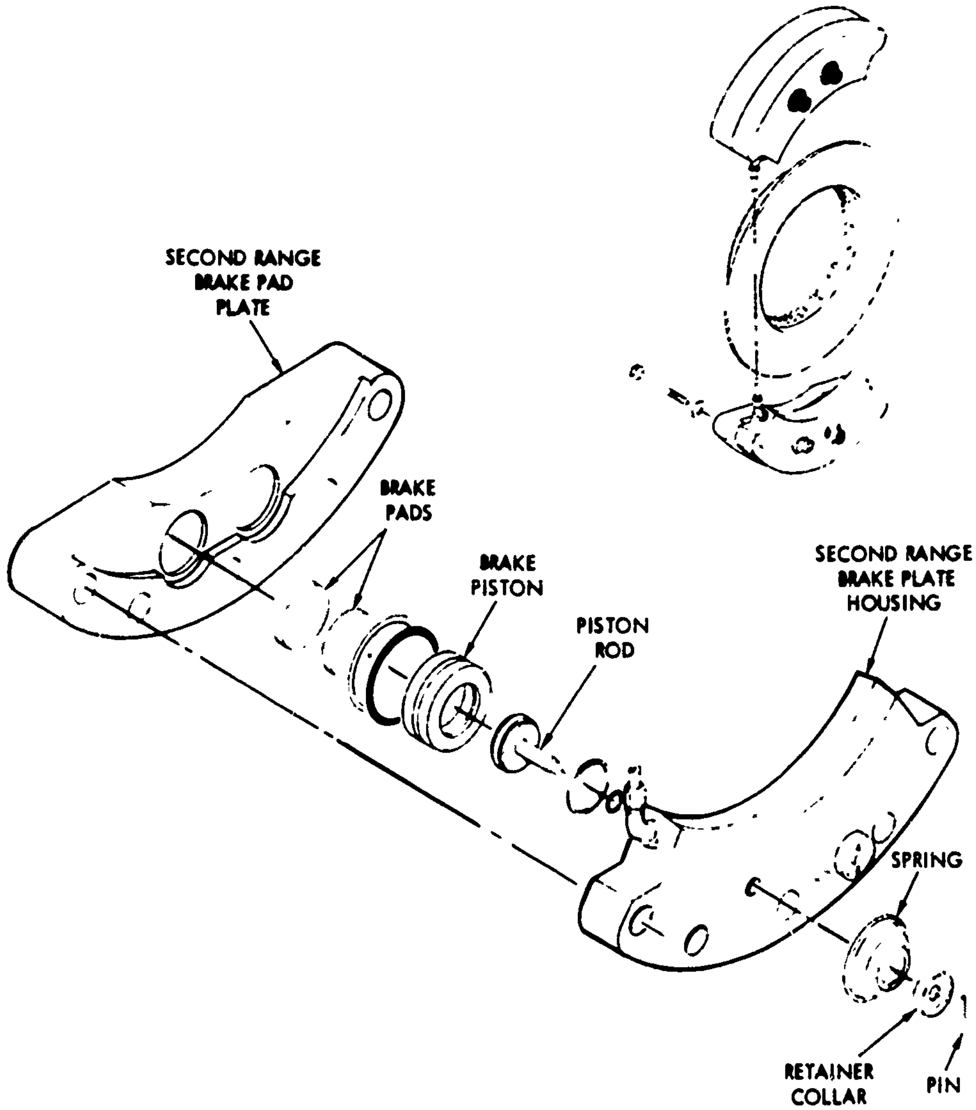


FIGURE 21 Second Range Brake Assembly

d. Third Range Clutch

The function of the third range clutch is to connect the engine-driven bevel gear cross shaft to the right-hand first range brake gear and thereby drive the main cross shaft at constant speed (relative to engine speed) during third range operation. This assembly is a multidisk rotating clutch (Figure 22), hydraulically-operated using low (priority) oil pressure. As previously described, both input and output speeds of the clutch are identical at the moment of engagement. Slippage between the disks is zero and wear problems negligible. The driving or bevel gear cross shaft side of the clutch has plain steel plates, while the output side has plates faced on both sides with Raybestos-Manhattan, Inc. Material Type 3672-3.

Operation is similar to the other range clutches. A pilot signal from the control operates a relay valve which is similar to those used in first and second range. However, instead of high pressure, the actuating pressure comes from the priority oil system (100 psig minimum). This actuation pressure moves a ring-shaped piston which forces the clutch plates into contact.

For rapid and positive clutch release, a ball check type vent valve is used at the outer periphery of the actuating piston. Normally, when the relay valve applies pressure to the clutch, flow past this ball and out the vent causes the ball to seat and holds it in this position while the clutch is applied. When the relay valve vents oil to the clutch, centrifugal force on the ball is greater than the pressure developed by centrifugal forces on the residual oil trapped

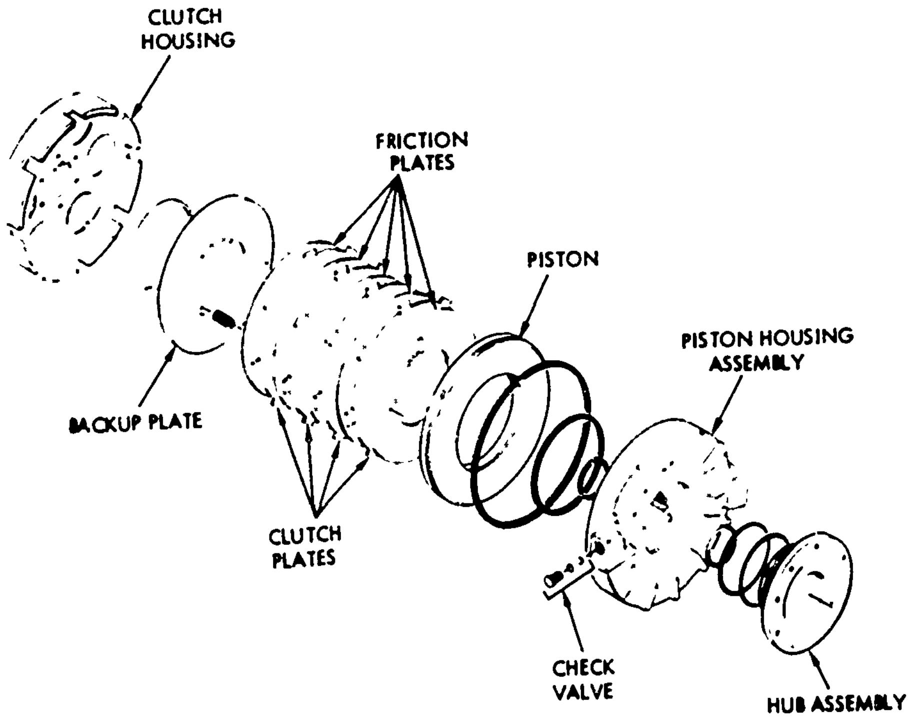


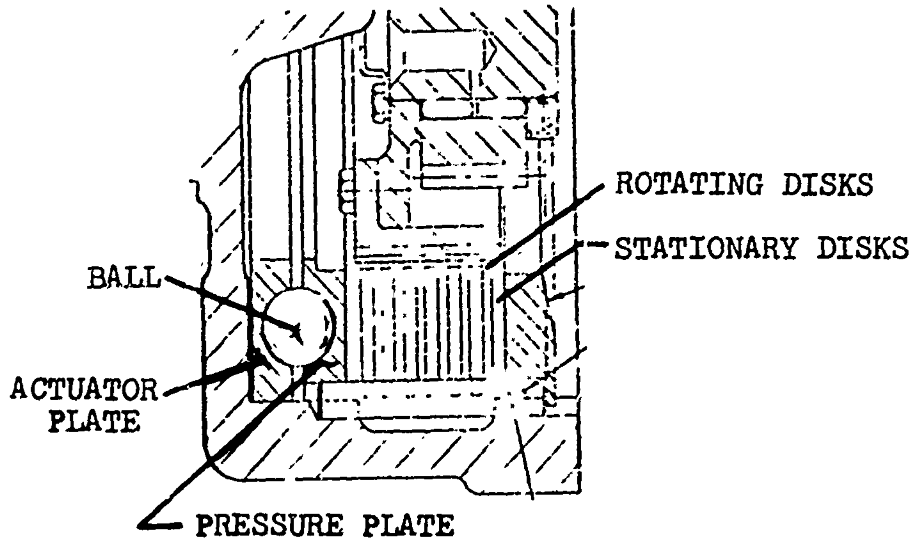
FIGURE 22 Third Range Clutch Assembly

in the clutch cylinder. The ball unseats and allows this residual oil to be exhausted from the cylinder and assures complete release of the clutch.

e. Service Brakes

Although not a functional part of the transmission propulsion or steering requirements, the vehicle service brakes (for emergency stopping or parking of the vehicle) are included within the transmission package for practical considerations. For the HMPT-500 the service brakes are mechanically-actuated, multiple disk types located in each output housing. They consist of six rotating disks and seven pressure (stationary) disks for a total of 12 contact surfaces. The stationary plates are faced with a sintered bronze material (Raybestos-Manhattan, Inc. W-1349). The rotating disks are splined to the output planetary carriers and the stationary disks are locked against rotation by detents in the outer periphery that engage pins recessed into the output housings. The entire brake assembly is fastened into the output housing by a retaining ring.

To actuate the brake a pressure plate is forced against the stacked disks. This pressure plate is moved axially by rotation of the actuating plate and by the action of eight balls that ride in tapered grooves in the pressure plate and in the matching actuator plate (Figure 23). The rotating plate is moved by a shaft and linkage mechanism in each output housing. An adjustment screw in each linkage adjusts the brake to establish and maintain proper brake pedal travel.



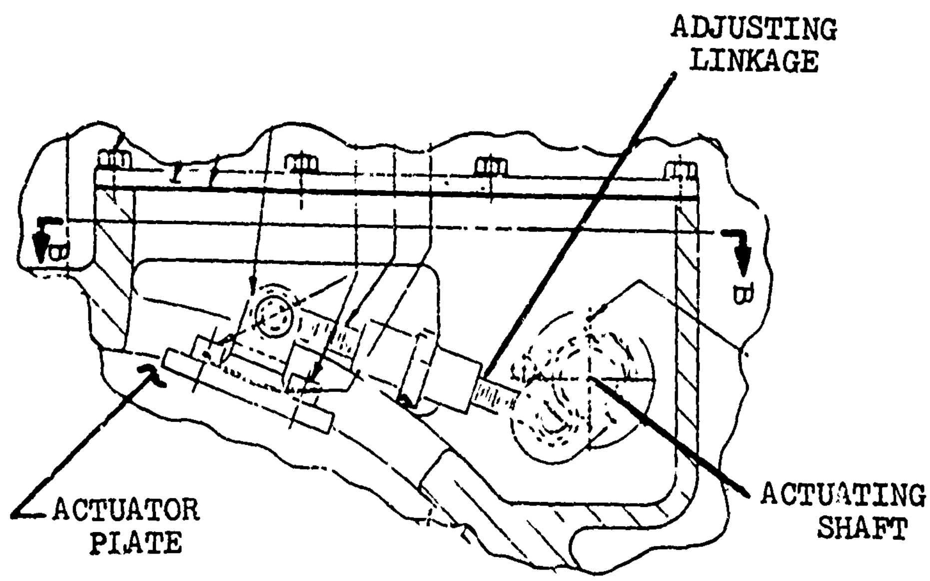


FIGURE 23 Service Brake Actuating Mechanism

Heat generated during braking is dissipated by cooling oil that is forced into the plates from the make-up oil system. Triggering of the oil flow is accomplished by a pilot valve in the right-hand brake actuating shaft. The brake coolant and time delay valves are housed in the same casting (Figure 24). Time delay is achieved by the orificed flow venting through plug. The coolant flow to the brakes is controlled by the spring-loaded coolant valve piston.

5. Bearings

In general, the bearings are of the antifriction type. Ball, plain roller, taper roller, and needle bearings are used on the various shafts and spindles throughout the transmission.

Taper roller bearings have been incorporated into the shafts that support the input spiral bevel gears where there are high radial and axial force components that make the taper bearings ideally suited for the application.

The five planetary gear sets (both outputs, both second range sets, and the steer differential) contain needle bearings within the planet gears. Since the space was limited, all of the planet gears required a full complement set of loose needles. In order to avoid helixing, two rows of needles were installed in all locations. Both the spindle and the gear bore are case carburized steel (either A151 4620 or 8620) hardened to 58 R_c minimum with a surface that is finished to a $\stackrel{16}{\checkmark}$, followed by a polish.

Except for the idler gear in the output of the second range gear train and the PTO idler gear, the power shafts and gears are supported by ball bearings. In each of the two idler gears, roller bearings (one in the second range gear

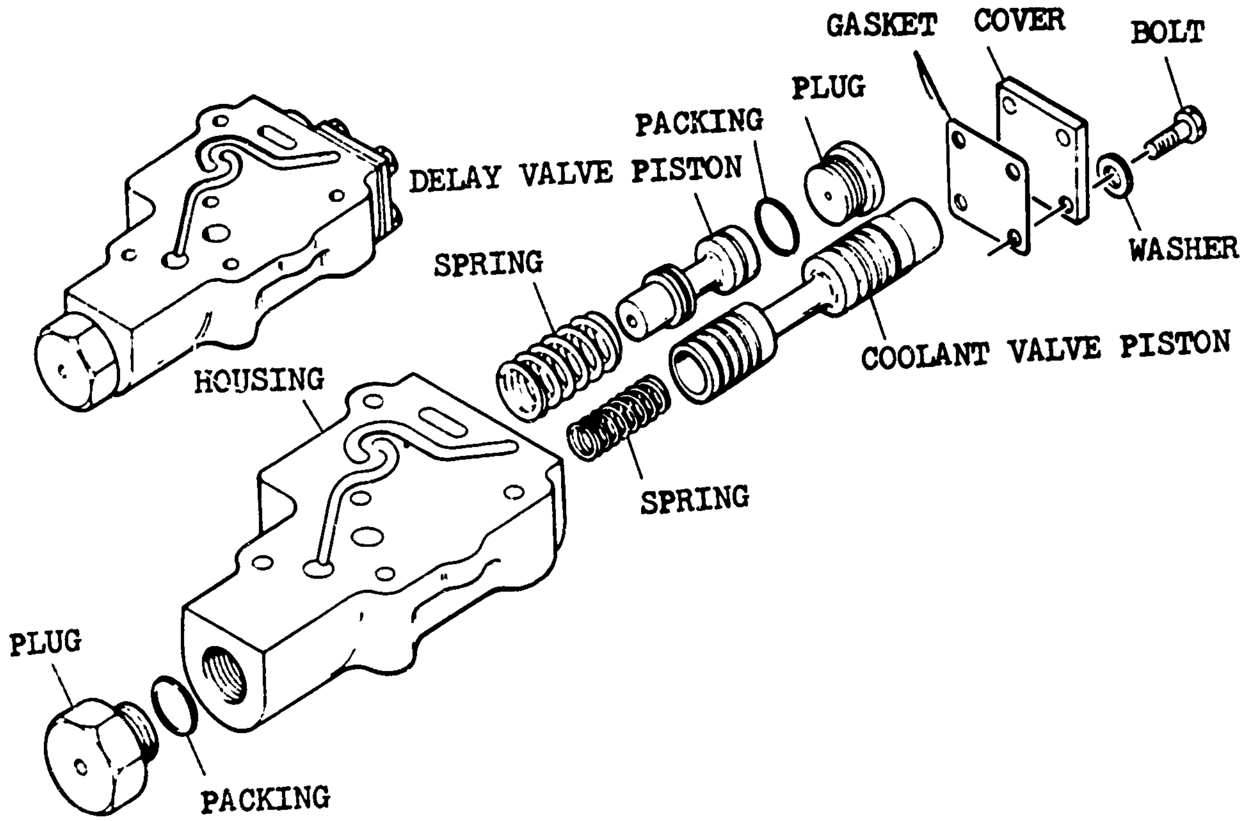


FIGURE 24 Brake Coolant and Time Delay Valve Assembly (exploded view)

train and two in the PTO idler gear) were incorporated to support the high loads that are frequently encountered.

The shafts on all of the gerotor-type pumps have journal bearings that are pressure lubricated by the discharge oil pressure from the pump.

The outboard end of the cylinder blocks are axially supported by a steel-backed bronze thrust bearing. Some of the splined shafts are also supported axially by thrust washers made from the same type of material.

6. Make-up Pressure and Lubrication System

The design of the make-up oil system is based very closely on the system used in the HMPT-100 2 steering transmission. It is comprised of a number of components and subassemblies spread through the transmission with the bottom of the main housing acting as the oil sump. Of these, the most complex is the make-up and auxiliary pump assembly (Figure 25) located in the lower section of the main housing.

Two cast housings (bottom and top housing) contain main and auxiliary pumps, pump drive shafts, pump drive spur gears, metering pump cover, auxiliary pump regulator and pilot valve, and a ball check valve. Each of these is described briefly below.

- Bottom Housing - an aluminum casting which forms the inlet or suction side of both large pumps. Provision is made to install a 20x20 mesh screen with an 0.034 inch opening over the suction inlet.
- Top Housing - aluminum housing with machined cavities for both large make-up and auxiliary pumps and the smaller metering pump. The flange

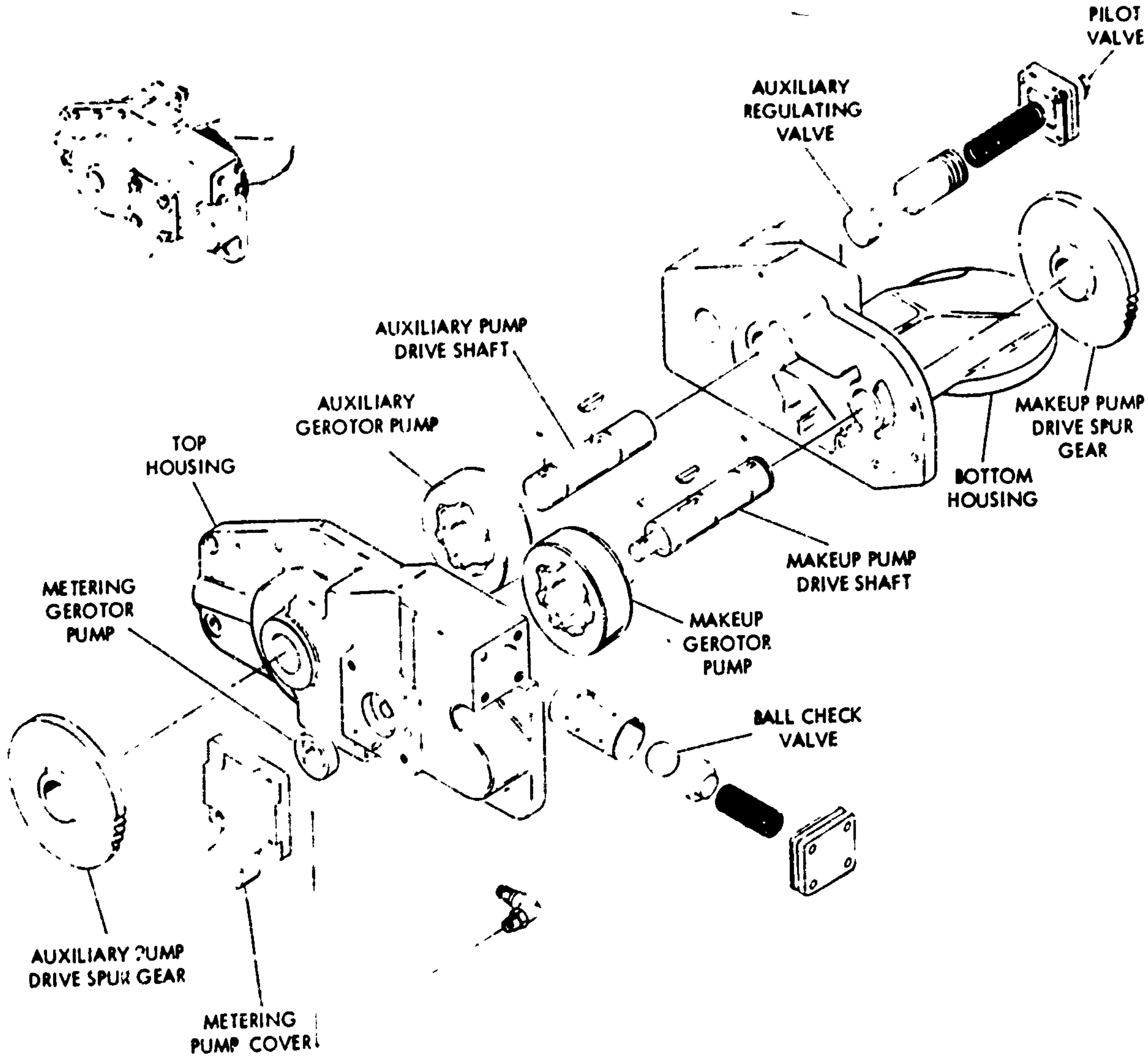


FIGURE 25 Make-up and Auxiliary Pump Assembly (exploded view)

on this housing provides a method for securing the entire pump assembly to the main transmission housing.

- Make-up and Auxiliary Pumps - commercial gerotor pumps as made by the W. H. Nichols Co. with a displacement of 3.6 cubic inches per revolution.
- Metering Pump - same as main pumps but with a displacement of .093 cubic inch per revolution.
- Pump Drive Shafts - two similar shafts made from 4140 steel, heat-treated to Rockwell C32-38 and sulfurized on all surfaces. The main make-up pump shaft is necked-down at one end to drive the metering pump.
- Make-up Pump Drive Gears - 60-tooth, 12 diametral pitch spur gears mounted on each drive shaft. The auxiliary pump drive gear meshes through an idler gear with the engine-driven side of the disconnect clutch. The main make-up pump drive gear meshes through an idler gear with a drive gear on the input bevel shaft. All three pumps are driven at 1.6 times engine speed.
- Auxiliary Pressure Regulating Pilot and Check Valve - this is a conventional pilot-operated regulating valve which, in conjunction with the check ball, becomes an unloading valve for those periods when the oil demands are low and the auxiliary oil supply is not needed. The regulating system is shown in Figure 26. As shown, the system is in the regulating mode with discharge from the auxiliary and tow pumps discharging through the check ball to the make-up system and the excess flow discharging to sump through the main auxiliary spool. If make-up pressure rises above the 120 psi regulating pressure of the systems,

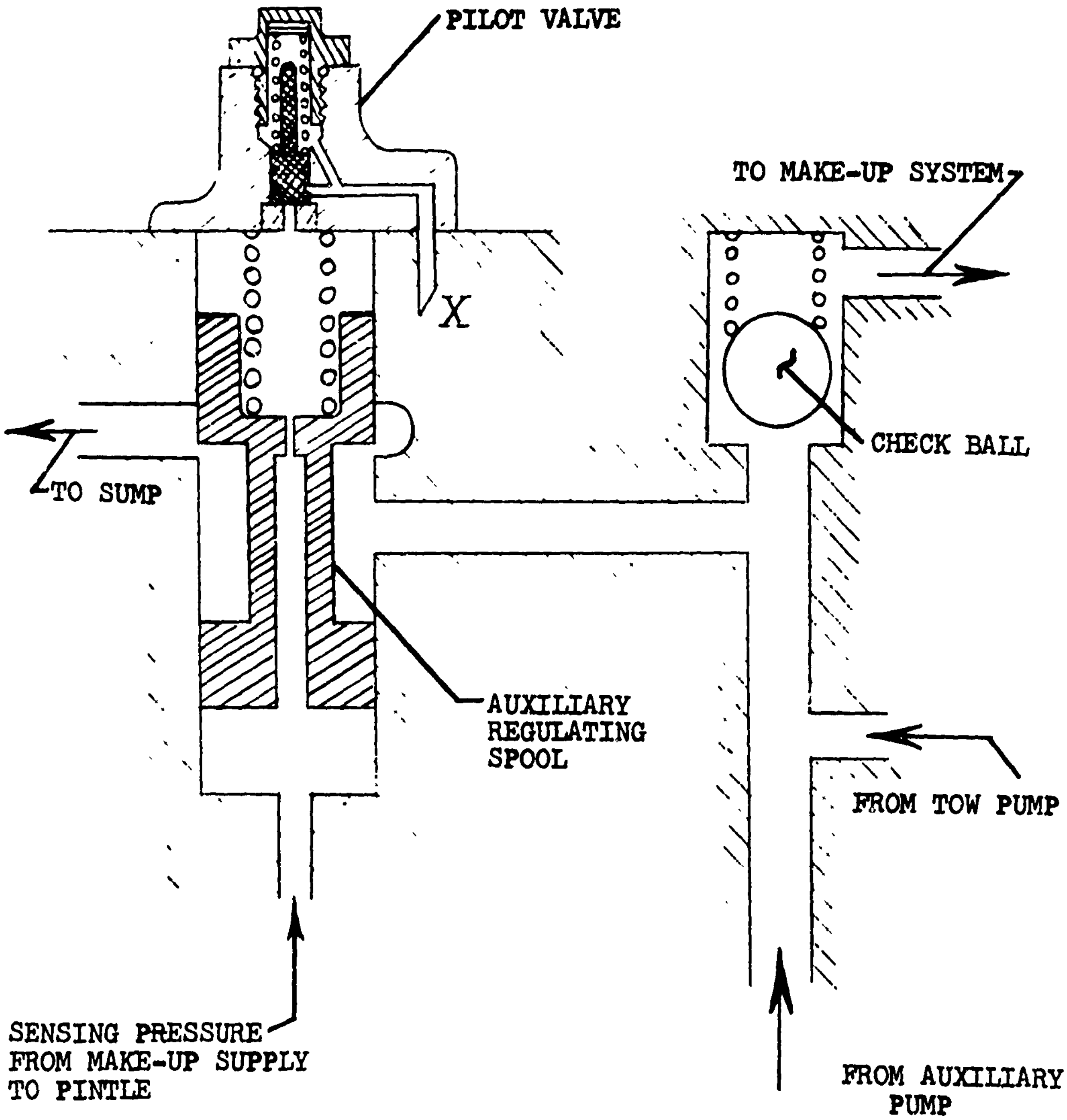


FIGURE 26 Auxiliary Pressure Regulating System

the check ball seats and all flow from both auxiliary and tow pumps is dumped to sump. In this mode both pumps are unloaded and pumping losses are minimized. Operation of the regulating valve is conventional. The pressure being sensed is applied to one end of a spring-loaded spool. An orifice through the spool applies a pressure signal to the opposite end of the spool. This pressure is applied to the spring-loaded pilot valve spool. If the pressure is not sufficient to move the pilot enough to vent flow back to sump, the pressure is the same on both ends of the large spool and the spring load keeps the spool seated at the sensing pressure end. If sensed pressure rises, the pilot valve spool is displaced until some flow is vented and a new pressure balance occurs across the main spool. As soon as the sensed pressure reaches the set value for the regulator, the main spool moves enough to vent all excess oil back to sump. Regulating pressure is established by shimming of the spring in the pilot valve.

This type of regulator has been used on all previous transmission designs and has excellent regulating characteristics over a wide range of flows and temperatures.

The remaining components in the make-up pressure and lubrication system are:

- Main Pressure Regulating Valve - this valve is identical to the auxiliary regulating valve except that the regulated pressure is adjusted to 150 psig. The main pressure regulator is located on the bottom right side of the transmission adjacent to the sump plate (Figure 27).
- Priority Valve - this is a pilot-operated regulating valve (Figure 28) similar to the main and auxiliary pressure regulators. It uses the same

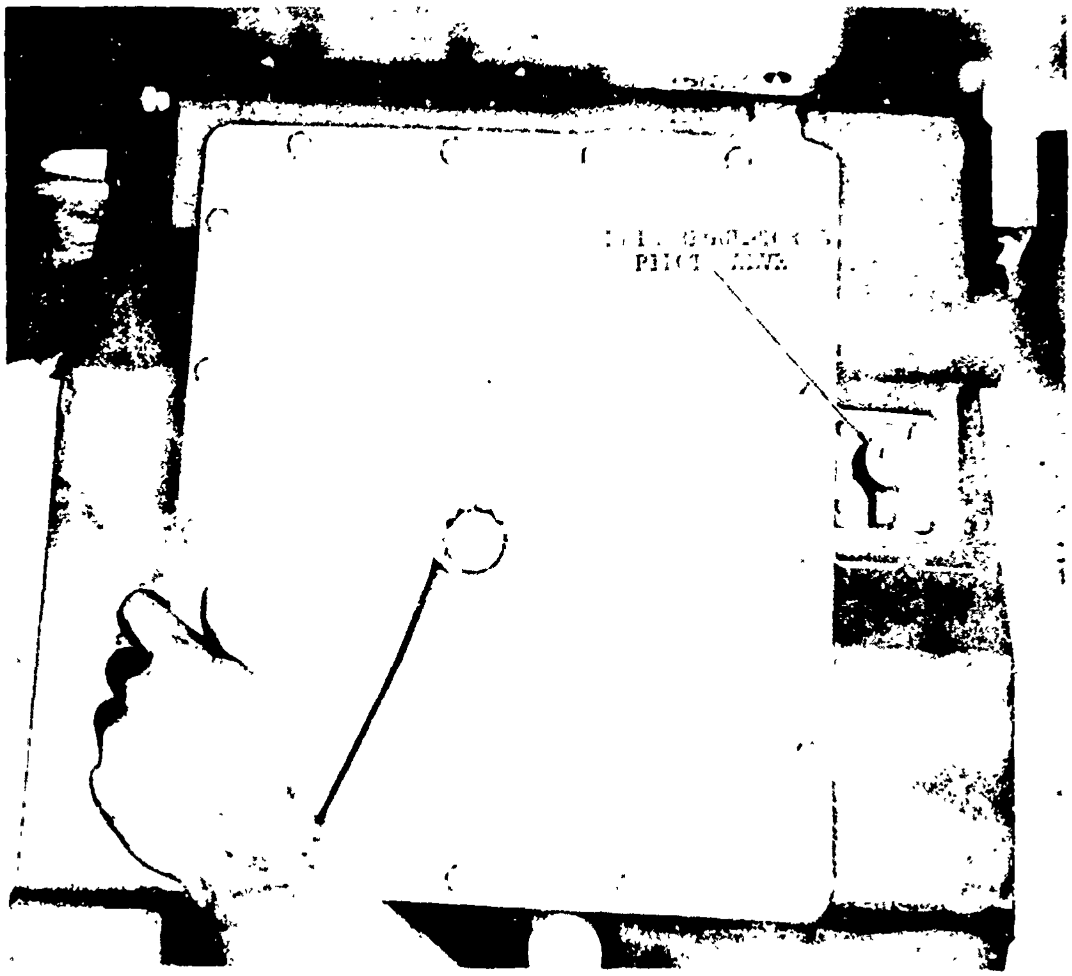


FIGURE 27 Location of Main Pressure Regulating Valve

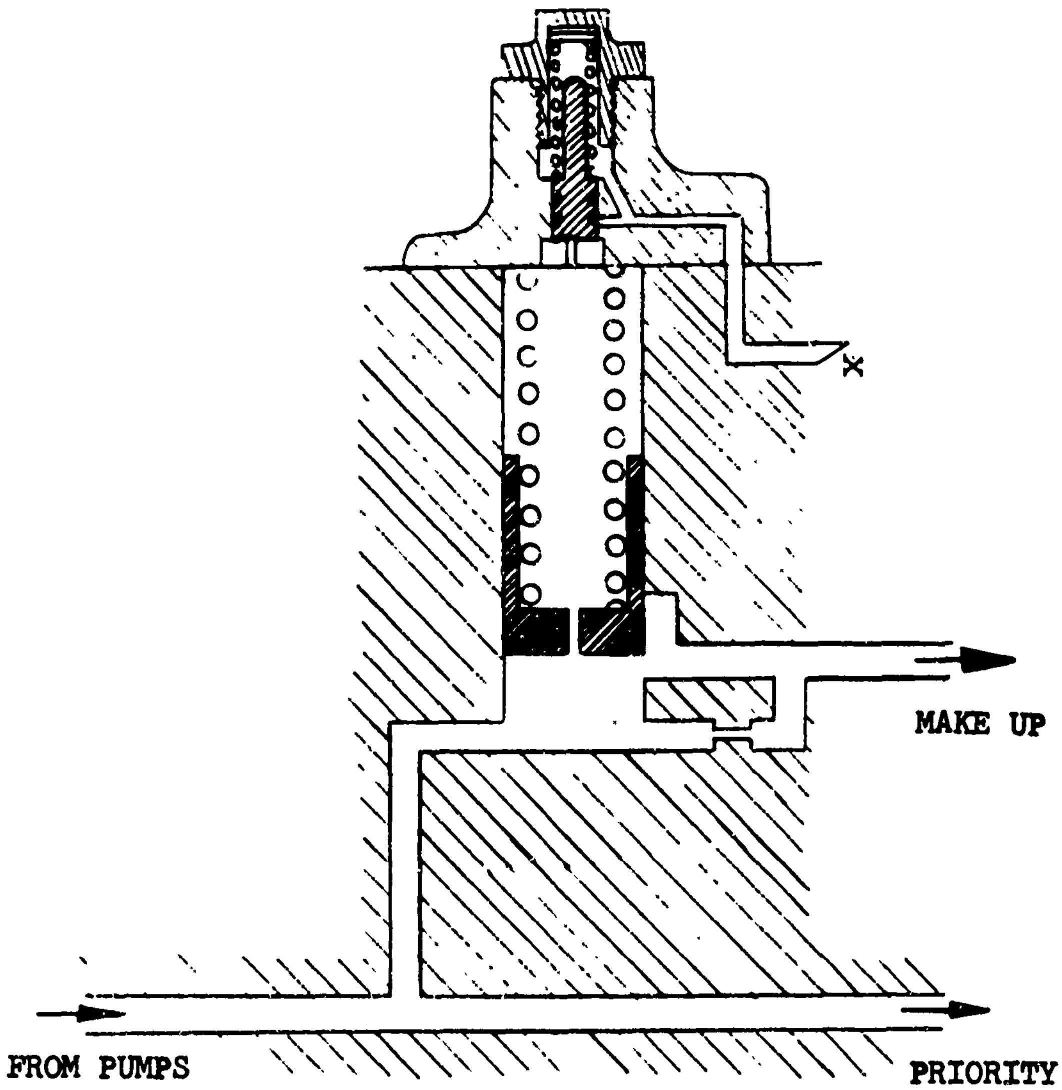


FIGURE 28 Priority Pressure Valve

pilot valve configuration, but the main spool is slightly larger in diameter than the other regulators. It also differs slightly in operation in that a by-pass orifice is used so that make-up flow can never be reduced to zero.

- Tow Pump - a fourth commercial gerotor pump, located in the lower right-hand main housing, supplies oil pressure for towing and push-start of the vehicle. The assembly (Figure 29) consists of two cast housings, the gerotor elements, a reversing shroud, a drive shaft, and the tow pump drive gear. Smaller than the make-up pump, this pump has a displacement of 1.7 cubic inches per revolution.
- Temperature Controller - a commercial thermostat (Figure 30) is used to regulate oil temperature. The thermostat also acts as a relief valve whenever restriction through the oil cooler exceeds 60 psi.
- Filter - a commercial 40 micron filter element (Figure 31) is installed just downstream of the temperature controller to filter all oil "entering" the transmission. A filter relief valve (Figure 32) is used to prevent collapse of the filter element if pressure surges are encountered such as during cold start-up.
- Lubrication System - Because of the large quantity of oil which is continually sprayed throughout the interior of the transmission (gear rotation and hydraulic unit leakage), every effort is made to use this free source of lubrication rather than to build in positive, pressurized lubrication. Where the part to be lubricated is adequately exposed to such spray, no special measures are taken. For other areas (e.g., input idler and output bearings) where spray does not reach the bearings, a catch tray and drip tube arrangement is utilized.

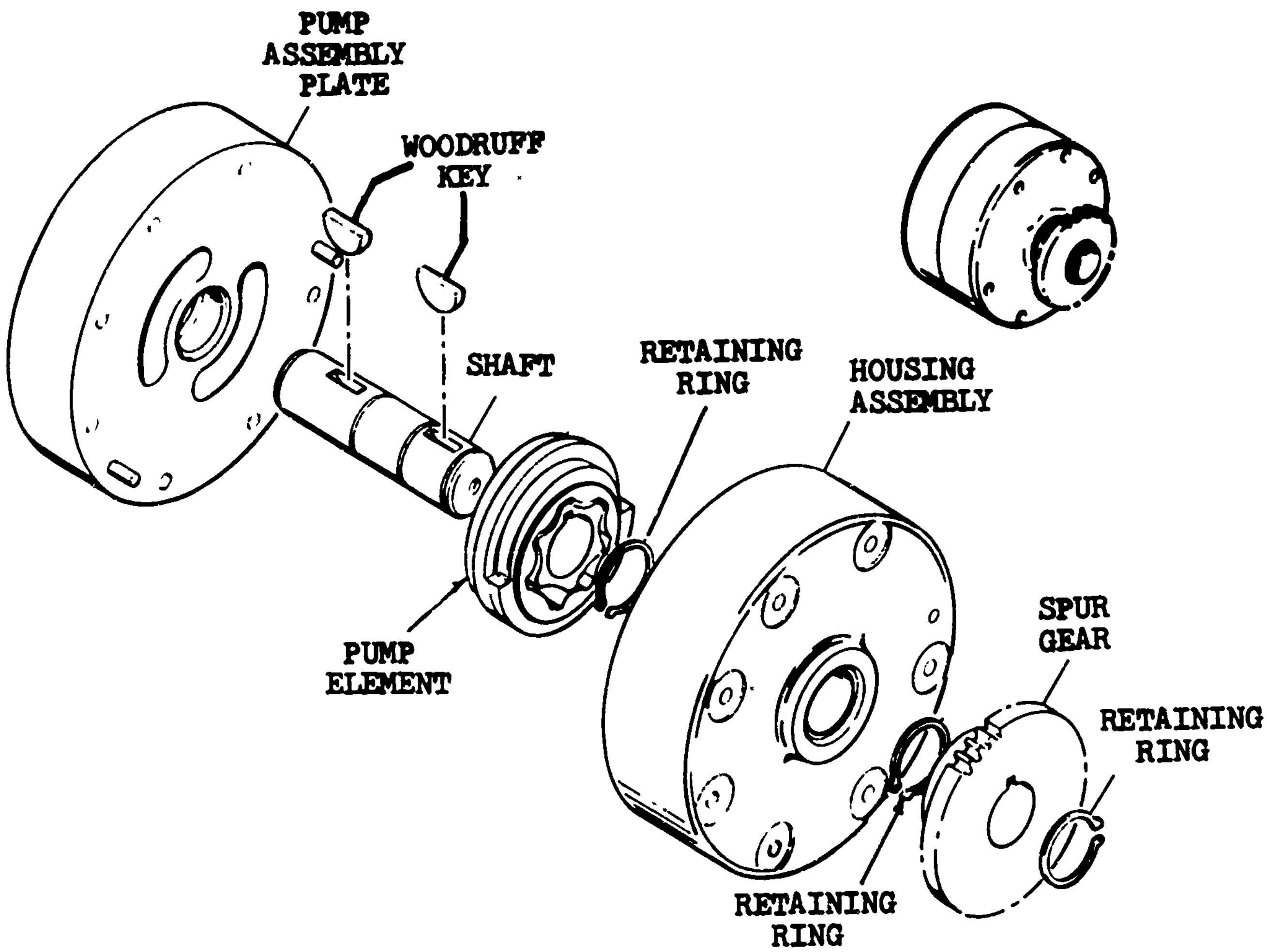


FIGURE 29 Tow Pump Assembly (exploded view)

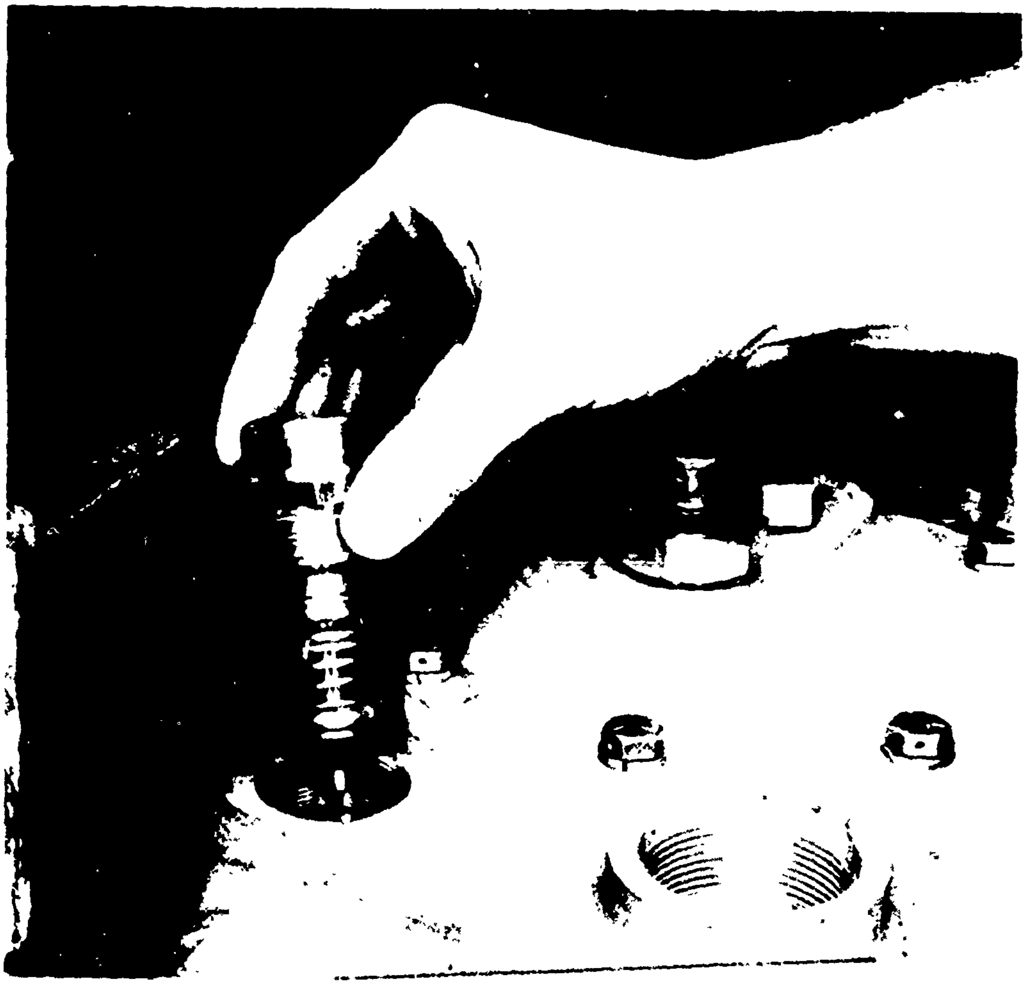


FIGURE 30 Thermostat Installation

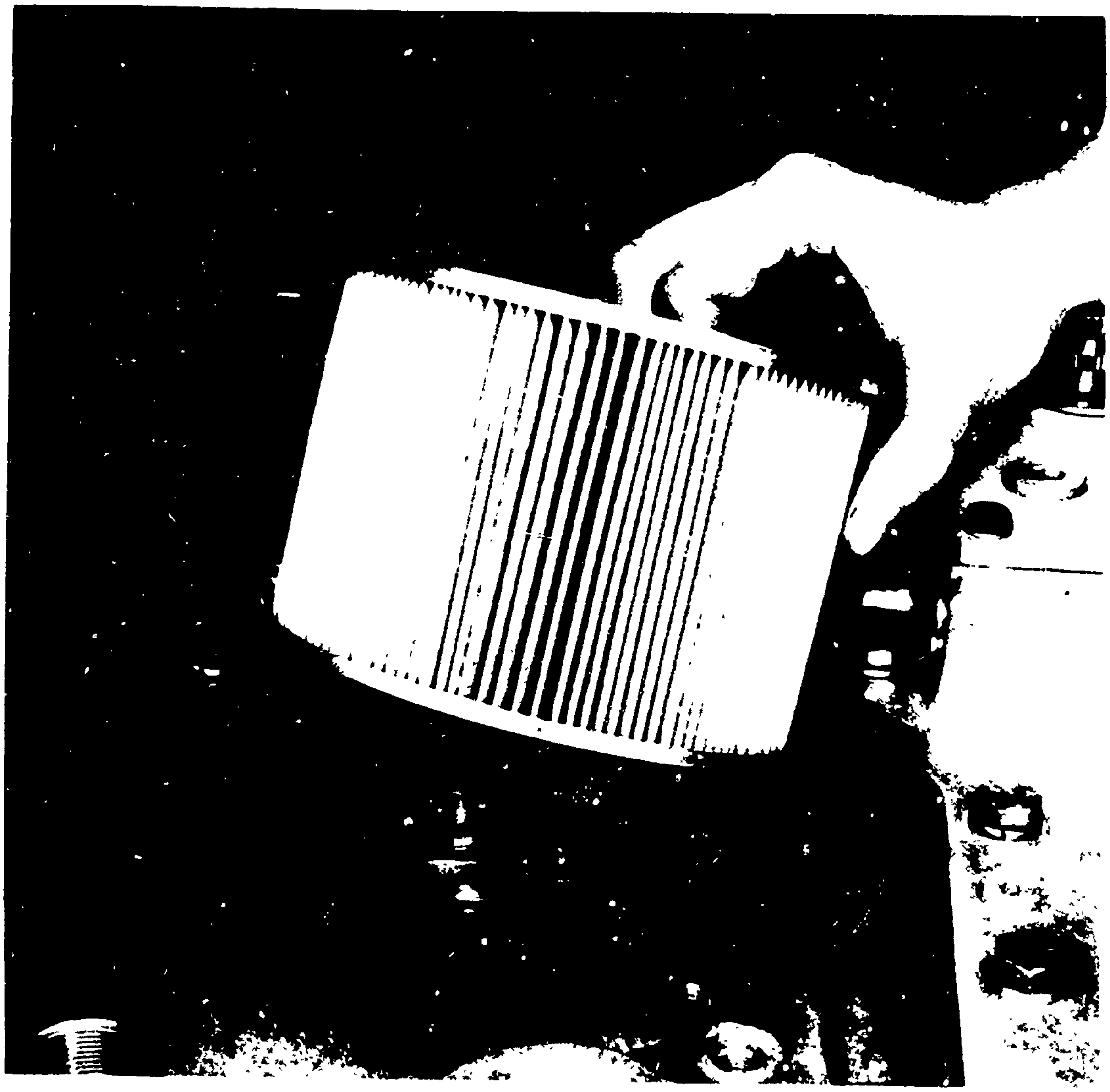


FIGURE 31 Filter Cartridge Installation

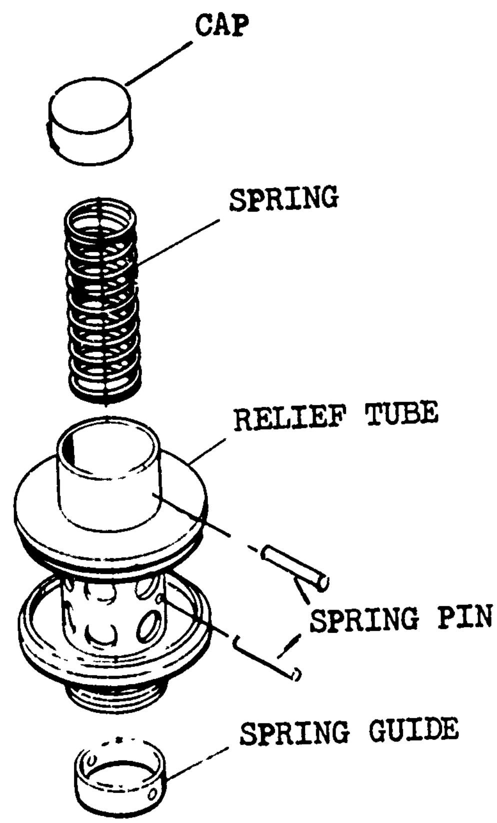


FIGURE 32 Filter Relief Valve Assembly (exploded view)

Finally, in those areas where experience has demonstrated the need for positive lubrication, the low pressure oil system is tapped to provide the necessary quantity and placement of oil. In this design, it was felt that only three areas would need such treatment: the cold start disconnect clutch plate area, the second range first planetary gear bearings, and the second range second planetary gear bearings. In each of these locations the initial design called for an orificed jet of make-up oil to be aimed into the path of the part requiring lubrication with final oil distribution accomplished by centrifugal action. A later discussion will describe modifications that were required to fully lubricate these and other critical areas.

7. Controller

The design of the HMPT-500 controller was undertaken with four primary objectives:

- Components from previous designs should be used wherever practical.
- External oil leakage should be eliminated by packaging all active hydraulic components beneath the controller plate.
- Operational adjustments should be simple, easily accessible, and should not be a source of external oil leakage.
- Pressure tap points should be readily accessible for trouble-shooting if problems occur.

The final controller design met these criteria in practically every respect and also met the restrictions posed by packaging dimensions and weight limitations.

The entire controller is built around a single aluminum cast plate which is flange to bolt to the main transmission housing (Figure 33). On the top of the controller are five shafts:

- Range selector
- Steer input
- Cold start clutch
- Accelerator pedal input
- Output to engine throttle.

In addition, there are three cover plates on the top for assembly, service, and test purposes. The long slender cover plate is needed for access to the pilot valve areas of the main actuators for connecting and disconnecting the control ratio/steer arm when installing or removing the control. A second small plate provides access to both the neutral adjustment and to the three adjustable clutch pilot valve actuator arms. The third cover plate is on the raised portion of the control (Figure 34) which contains the engine scheduling cams, the throttle knockdown linkage, and the maximum throttle input limiter adjustment.

All of the remaining components are located below the plate (Figure 35). The primary hydraulic components, except for the three range change pilot valves and the steer governor piston, are incorporated in a single housing. Included are the main governor assembly, the range selector valve, the power piston, the cold start clutch valve, the clutch pilot feed valve, the cross shaft interlock valve, and the steer governor valve. The three range change pilot valves and a steer upshift inhibitor, which prevents a 1-to-2 range change if steer exceeds a preset value, are located in a separate housing adjacent to the main ratio control cam. The remainder of the linkage is

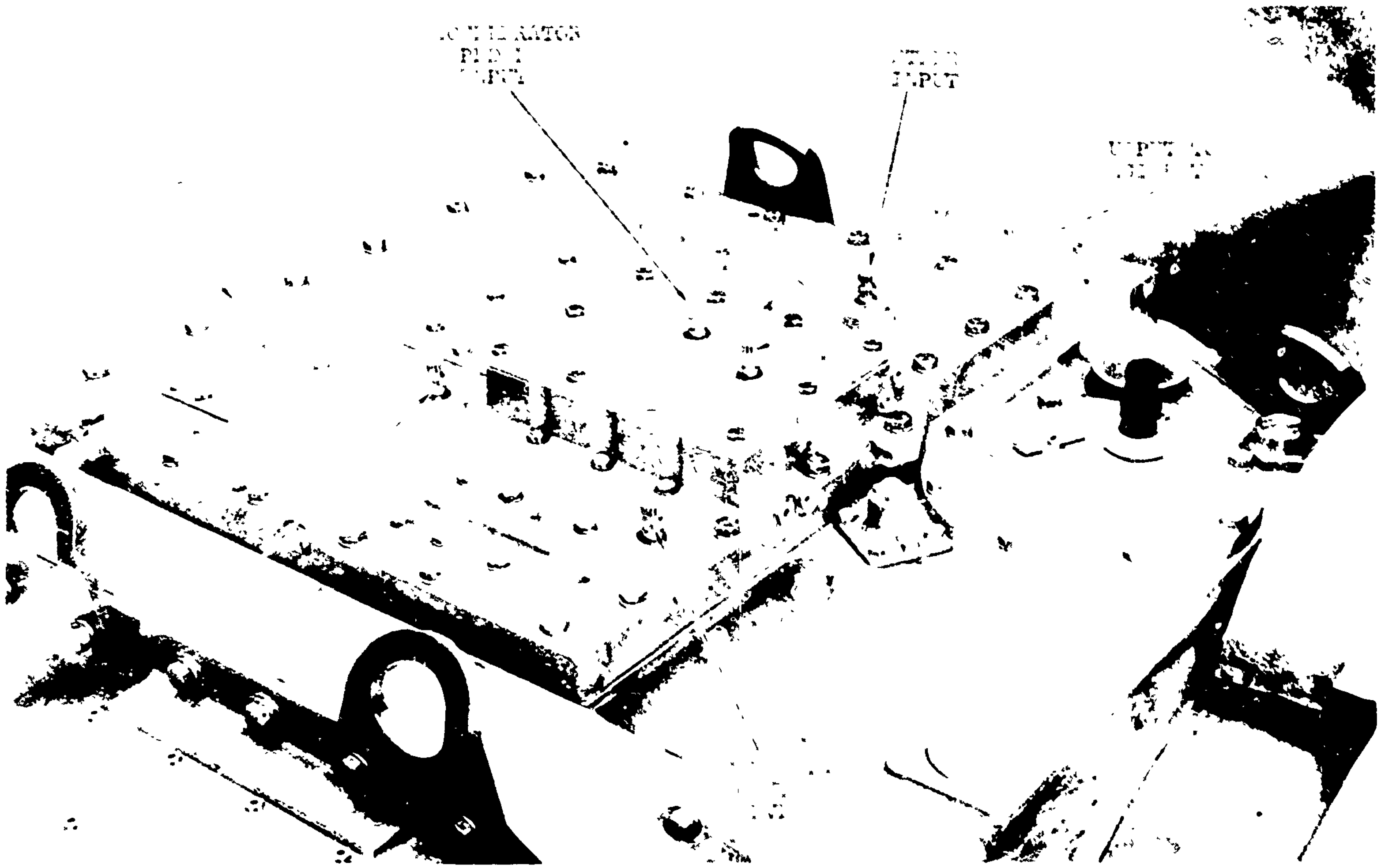


FIGURE 33 Controller Installation

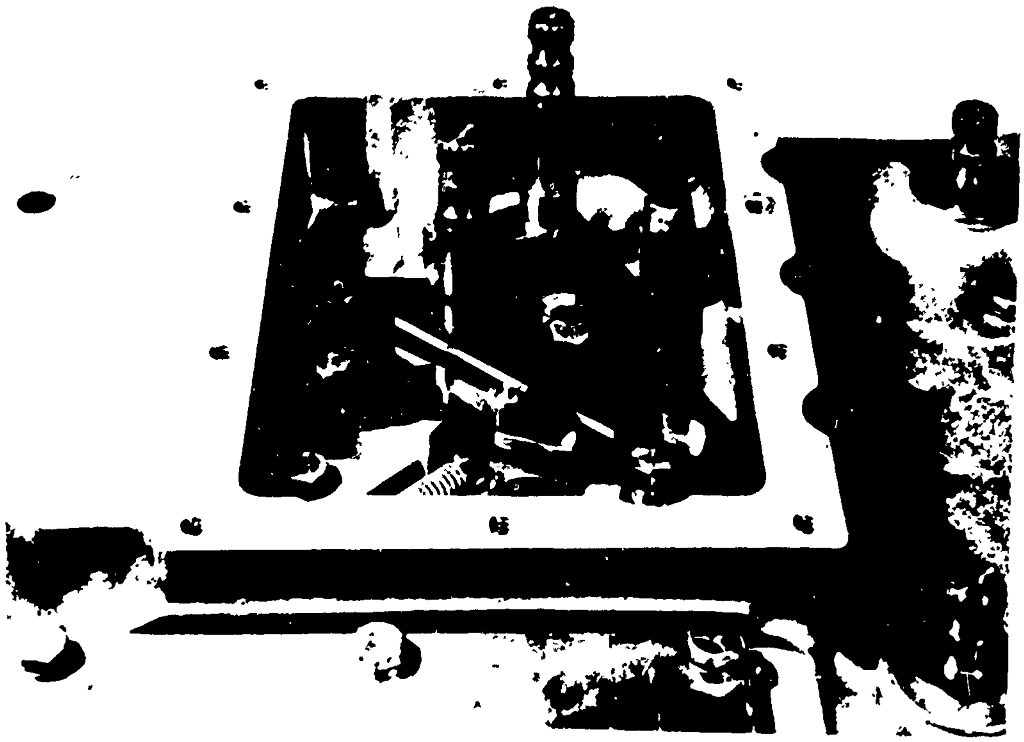


FIGURE 34 Engine Scheduling Cams and Linkages

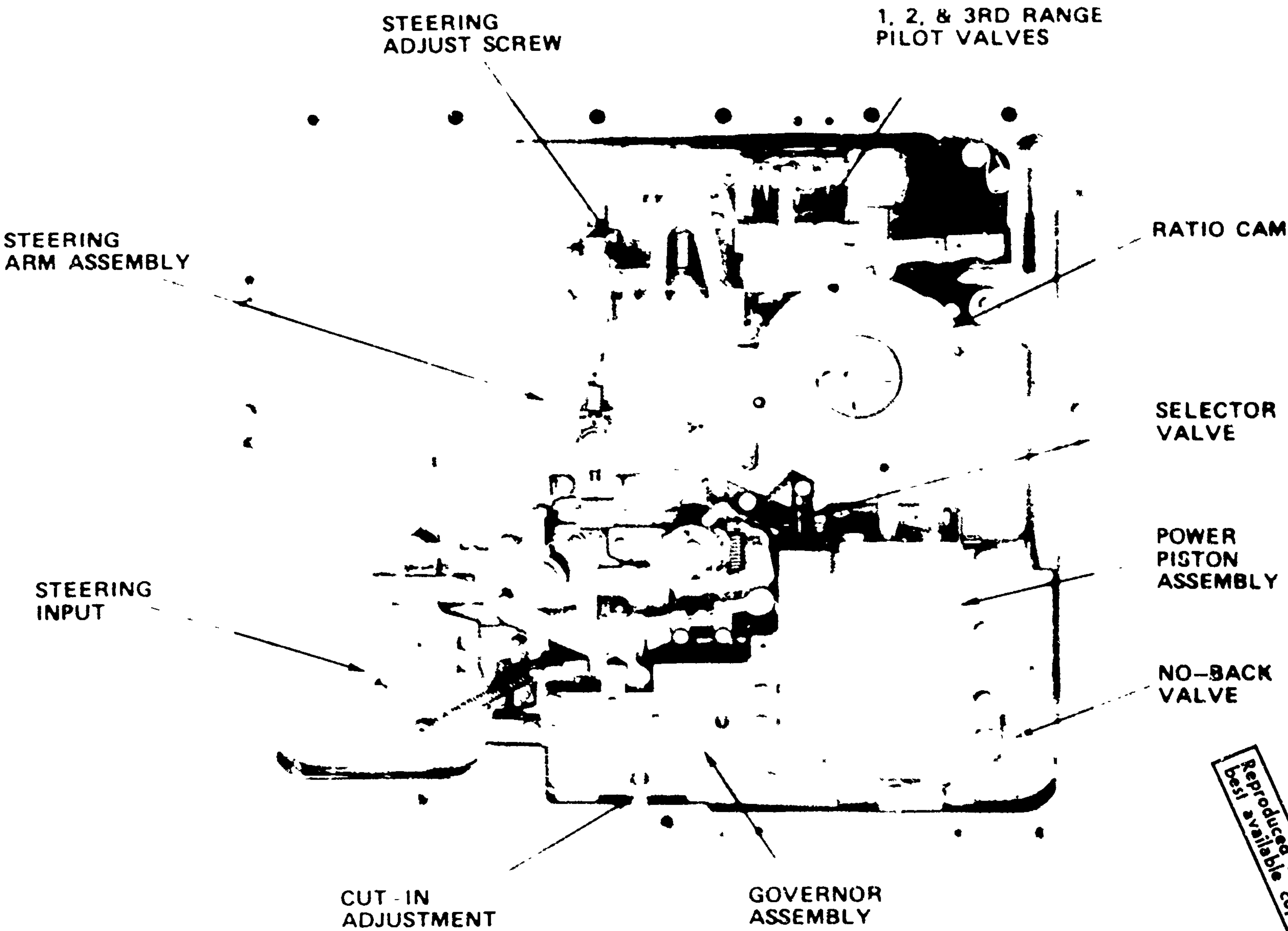


FIGURE 35 Controller Assembly - Bottom View

fastened directly to the control plate. Figure 36 is a hydraulic-mechanical schematic of the primary control assembly. Not only are the hydraulic-mechanical components and connections shown, but also the overall layout has been made such that it may be referenced directly to the hardware arrangement (see Figure 35).

In keeping with the stated design goals, the hardware design of this control was based closely on that of the HMPT-100-2 control. Where possible, identical components were used. In cases where other factors were overriding, the components were, if not identical, at least of very similar constructions. Construction details of the various areas are discussed briefly in the following paragraphs and changes from previous hardware designs are noted.

a. Main Governor

This unit is practically identical to previous designs. Speed sensing spool and sleeve, pilot valve spool and sleeve, summing bar, springs, and linkage are existing parts. The major change is in the cut-in adjustment which has been changed to a star wheel type of mechanism because of inaccessibility for the conventional screwdriver slot type of adjustment.

b. Selector Spool

This is an aluminum hardcoated spool, the same as in the HMPT-100-2 except that the detent grooves on the spool have been eliminated and the detents designed into the selector shaft components. The change was required primarily to reduce overall length of the spool. It may be noted in Figure 36 that two orifices are drilled into the hydraulic feed passages immediately adjacent to the selector spool.

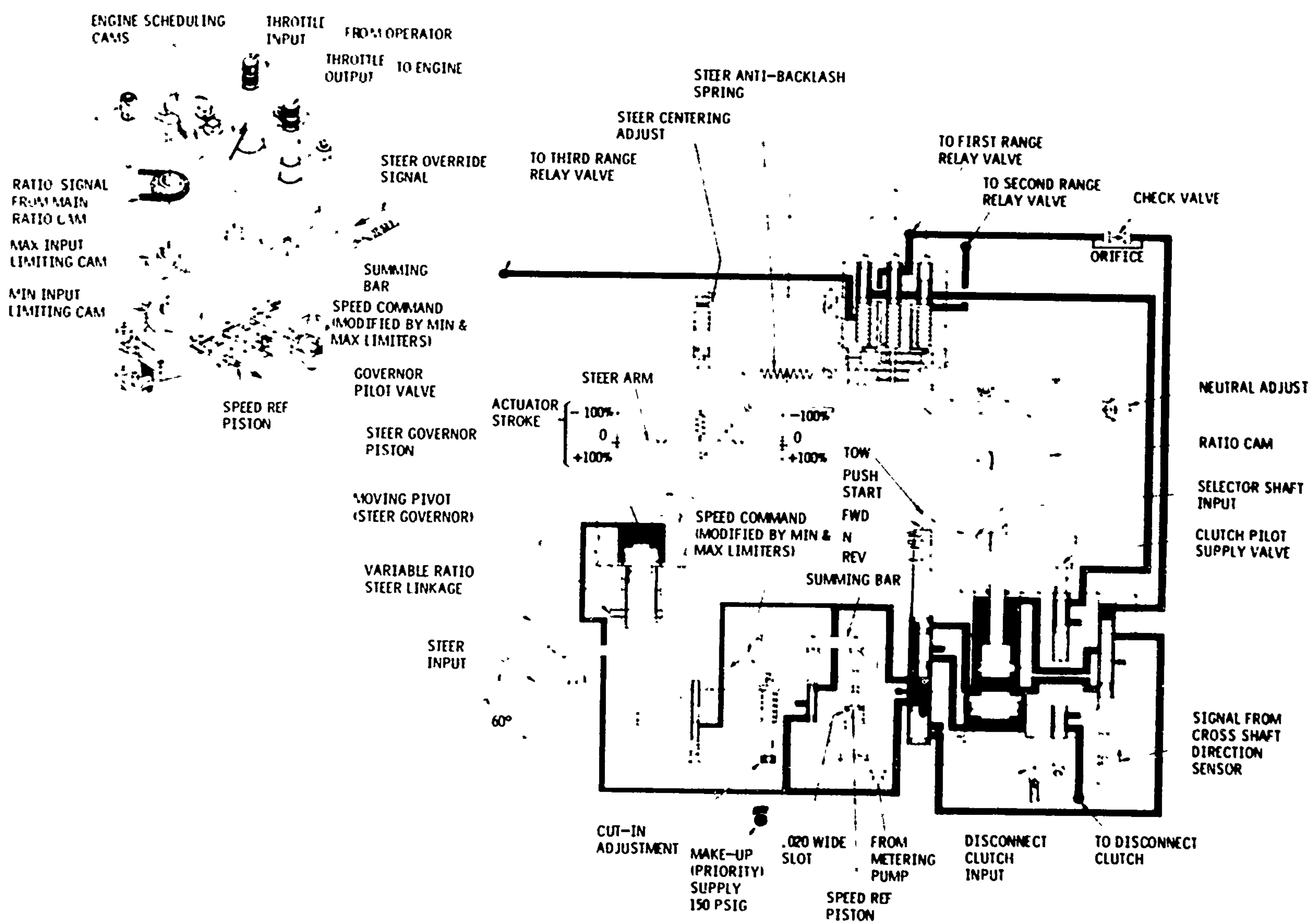


FIGURE 36 Controller Hydraulic Schematic

One of these allows the selector valve to be moved to neutral at any time from either forward or reverse without damage or unreasonable vehicle deceleration rates. The second orifice is required when shifting rapidly from forward to reverse under power to rock the vehicle (see Work Statement, Appendix A).

c. Power Piston

This is identical in function to previous designs but the hardware has been modified to minimize length requirements. Previously the gear rack was mounted in the center of a long double-ended piston; the new design uses a small piston and piston rod to which the gear rack is attached. With this arrangement the schedule cam shaft and drive gear are mounted remotely from the power piston housing. The reverse piston is a short floating piston, one-quarter inch in diameter larger than the forward piston.

d. Miscellaneous Spool Valves

Of the remaining four spools, the disconnect clutch valve spool is taken from the HMPT-100-2 hardware, while three are designed specifically for this job.

e. Steer System

With the exception of the steer governor piston and some of the steer/ratio arm parts, the components are all new designs but bear a marked resemblance to existing designs. The variable ratio linkage has been modified, for manufacturing convenience, to use parallel rather than angled slots. However, input-output relations are essentially identical.

The steer/ratio arm differs only in width across the pins due to actuator spacing. Of the remaining parts, the main steer shaft, steer centering adjustment, steer shaft housing, and connecting links required some modification because of space requirements but they contain the basic elements and appearance of former designs.

f. Main Ratio Cam and Cam Follower

The main ratio cam is a closed track face cam with a groove contoured to give the stroking schedule of Figure 7. Because of a three-to-one ratio between output end and follower in the cam follower arm, the vertical displacement in the cam track is one-third of the actuator displacement. At neutral and both shift points, a flat is introduced into the cam track. At neutral this assures a fixed and definite neutral position which is not affected by small amounts of wear in linkage or other parts. At the shift point the flat permits a momentary pause in the stroking schedule as the clutch pressures are interchanged.

On the outside of the cam are three steps or levels corresponding to the three ranges. Location of the steps is oriented to the cam track flats so that shifting occurs during the pause in actuator stroking. Material of the cam is SAE 4615 steel. The closed track and the outer cam face are case hardened to Rockwell C58 minimum.

The cam is fastened to a hub by three screws which allow the cam flat at neutral to be referenced to the power piston neutral at assembly. The hub, in turn, is keyed to the shaft which is rotated by action of the power piston.

One end of the cam follower arm is anchored to the control plate through an eccentric pin in the plate. Rotation of this pin adjusts neutral creep of the vehicle. The other end of the cam follower arm is attached to the steer/ratio arm. A typical override link in the cam follower arm ensures against damage to the control linkage, particularly if gross relative displacement between actuators and control occurs during nonoperating periods (vehicle shipment).

g. Clutch Pilot Valve Assembly

This consists of a housing with three clutch pilot (spool) valves, three adjustable cam follower arms, and the steer upshift inhibitor spool. It was recognized early in the design that clutch timing would be a critical item in assuring smooth range changes under a variety of loads. Thus, the initial design has been made with adjustable cam follower arms to permit varying clutch timing (overlap, underlap, etc.). Once an acceptable combination for both range change conditions is achieved, the adjustable feature will be modified to use a single fixed arm assembly.

Since all three pilot valves move on each range change, it can be seen that one pilot valve performs no function during each range change. For example, in the 1-to-2 range change, first range pilot vents pressure, second range pilot applies pressure to its relay valve, the third range pilot moves but does not port pressure in any manner. In the 2-to-3 range change, the first range pilot valve movement performs no function.

The steer upshift inhibitor valve is connected by linkage to the main steer shaft. Normally in low range the first range pilot pressure is applied through a 0.030-inch diameter orifice to the back side of the first range pilot valve. With no steer, the pressure is vented to sump through the upshift inhibitor spool. If a steer signal exceeding about 20° at the transmission input is applied, the vent passage is closed and full priority pressure builds up behind the first range pilot spool. The pilot spool diameters have been sized such that this pressure is sufficient to prevent the power piston from rotating the main cam when the cam step is forced against the first range pilot valve cam follower. If steer is reduced, venting of the pressure occurs and the upshift will take place if the control so commands.

h. Engine Scheduling Cams

One inherent advantage in this transmission is the ability to match engine speed and power to follow any desired power curve. Because of the vastly different throttle characteristics of products of various engine manufacturers, the control must have provision for matching the transmission to different engines. Physically, this is done with two cams located on the accelerator pedal input shaft just below the cover plate. Both cams are adjustable. The topmost cam adjusts cut-in and compensates for slight engine throttle differences near the engine idling point. The lower cam trims the top governed speed to match the desired maximum engine speed. Other than trimming the extremes of speed, there is no built-in adjustment for varying the engine-power schedule. If this is desired, cams with different contours must be installed.

i. Maximum Governor Input Limiter

For rapid accelerations when the output ratio is near neutral, an inhibitor on speed command to the governor is required to prevent undesirable vehicle reaction. This is accomplished by limiting the amount of acceptable command at neutral and, as ratio moves away from neutral (forward or reverse), gradually removing the restriction. If space were available, the main ratio cam could be used for the ratio reference. However, because of space requirements, a second shaft is rotated by a small chain drive at the same speed as the main cam shaft. A cam on this shaft is contoured to move a link which in turn will limit throttle input to the control governor in the low ratio ranges. The link is adjustable to vary the amount of restriction depending on vehicle-engine power combinations.

Typically the restriction is set to limit governor input command at neutral to 1800-2100 rpm and to remove all restrictions before the 1-to-2 range change point is reached. To permit full power pivot turns (ratio at neutral), the inhibitor is interlocked with the steer signal so that all restriction is removed whenever steer input to the control exceeds 25^o (full steer equals 60^o).

j. Throttle Knockdown Linkage

In designing a transmission for use with an unknown engine, it is prudent to provide a means to adapt to an engine with greater than the nominal design horsepower. Thus, if the HMPT-500 were to be used with a 600 input horsepower engine instead of the designed 500 horsepower, a simple linkage system has been built into the controller to accomplish this adaptation. If the transmission were to be used exclusively with a lower horsepower engine, the control could be simplified to eliminate such excess capability.

The linkage consists of a cam rotating on the same shaft as the maximum governor input limiter cam, a cam follower, and a linkage system which will reduce throttle by whatever schedule is contoured into the cam. For example, if it were desired to use a 600 horsepower engine but limit horsepower to 500 horsepower from neutral to the 1-to-2 range change and then increase available horsepower for top speed performance, the cam could be contoured for such a schedule. Any other specified schedule could be as easily contoured into the cam if it were considered desirable.

The linkage is so devised that the percent throttle reduction is the same at any fixed ratio point. For example, at full acceleration the engine throttle is reduced by 20 percent to its 80-percent position; at half accelerator pedal input the engine throttle would also be reduced by 20 percent to its 40-percent position.

k. Cross Shaft Interlock

There exists within the transmission one condition which the control governor cannot sense through its normal engine speed input system. Under ideal conditions the 2-to-1 downshift should always occur when the main transmission cross shaft has slowed and reached zero. With high road load, high engine power conditions, this zero cross shaft speed point is reached at a ratio considerably before the cammed point in the controller is reached. Reversed rotation of the cross shaft can then occur due to high leakage in the main hydraulic units. A device has been incorporated on the cross shaft which will sense the direction of rotation and, if reverse rotation starts to occur, signals the controller to reduce ratio rapidly and initiates a 2-to-1 downshift.

The mechanism consists of two ball bearings on the cross shaft which have been slightly preloaded against an outer ring. A two-link system extends from the outer ring to the cross shaft interlock spool in the controller. If while in second range the shaft rotation reverses, the interlock spool is tripped and a downshift occurs as the control is forced to a lower ratio. In first range the interlock spool is reset by applying first range clutch pilot pressure to the end of the spool. In second and third range, forward rotation of the cross shaft applies a slight load on the spool to keep it in its normal or "ready" position. The preload on the two bearings is maintained at such a low level that continued high speed forward rotation of the cross shaft does not cause bearing damage.

Thus, from the foregoing it may be seen that the controller is really a simple hydromechanical system that is relatively independent of the horsepower rating of the transmission. As such, it is amenable to design from standardized components where space conditions permit. It also has great flexibility in design in that changes or modifications for special applications can be readily incorporated into the basic concept.

8. Housings

The seven primary housings range in complexity from a simple cast plate for the controller to the 200-pound main housing casting requiring about 37 cores. In addition, a number of smaller castings are required for make-up pump housings, controller housings, and valve housings. For the controller plate and controller hydraulic housings, material is 356-T6 aluminum alloy since soundness is a prime requirement with strength of secondary importance. All of the remaining castings use 355-T6 aluminum alloy to take advantage of the increased strength at operating temperature (200°F-300°F).

From a casting standpoint none of the housings are considered to be extremely complex or to present any unusual characteristics which should make them difficult to procure. In fact, the castings for this transmission are felt to be less complex than those of its predecessor, the HMPT-100-2.

C. Fabrication and Assembly

Procurement of hardware for this program was a mixture of in-house machining and outside vendor fabrication, depending primarily on the types and prior commitments of in-house machines. All assembly of hardware was done in-house.

Raw aluminum castings, except the main housing, were procured from outside vendors. The main housing was cast at the Erie Foundry of the General Electric Company. All seven of the primary housings were set up for machining on the tape-controlled machines of the General Electric Ordnance Systems machine shops. Other small aluminum castings were machined by outside vendors.

Gears and all splined parts were fabricated at a number of vendors specializing in this type of hardware. The remainder of the machining on both specialized parts (hydraulic units) and normal small parts was an equal mix of in-house and outside vendor machining.

Initial transmission assembly met with the usual number of problems associated with all development programs that operate on accelerated schedules. Discrepancies resulting from both design and fabrication errors needed correction at each stage of assembly. However, no major errors were uncovered. Assembly of the first unit, SN 10, began on 8 January 1973 and was completed on 9 February 1973. This unit was scheduled to make initial evaluation tests and to be run on the dynamometer for 400 hours. The second unit, SN 11, was started through assembly on 12 February 1973 and completed assembly on 17 March 1973. It was planned that SN 11 would be installed in a vehicle and operated for 10,000 miles of durability testing. Unit number 3, SN 12, was to be maintained as a spare.

D. Testing

1. Initial Dynamometer Testing

When the initial assembly, SN 10, became available in early February 1973, the dynamometer test facility at Ordnance Systems was still under construction. However, a temporary facility which used a 6V53 Detroit Diesel engine was available for spin testing (no output load). On 9 February 1973, first spin testing began and was continued until 8 March 1973. During this period the transmission was torn down five times for examination and correction of minor problems. These included:

- clutches failing to release completely because of insufficient clearance between pistons and bores,

- minor rubs between rotating parts and castings,
- color indications on planet carrier spindles, bearings, thrust washers, and pinions.

Primary rework as a result of the spin testing was to increase lubricating oil supplies to all second range and output planet carrier spindles. It must be remembered, however, that all of this testing was without load at the outputs and, consequently, the gear trains had all operated without normal loading.

In early March 1973, the dynamometer facility became operational. Following the fifth spin test series, the transmission was given a major teardown and inspection and was reassembled for its first dynamometer test. It was installed in the test cell on 22 March and, despite the attendant difficulties of running both a new transmission and a new test facility, a green run (scan of normal power-speed spectrum) was completed on the following day.

Between 23 March 1973 and 8 May 1973 testing continued with two primary objectives:

- Define and correct all problems prior to the start of durability testing.
- Establish performance characteristics and compare to predicted values.

During the above test period the transmission was removed from the test cell and torn down eight times, partially on a scheduled basis and partially due to problems that occurred.

The problems that were found repeated, in general, those that had been noted during the original spin testing, namely, lubrication problems in the second range and output planet carrier needle bearing areas aggravated by the higher power loading on the dynamometer. Corrective action resulted in an increase of forced lubrication to all critical points to assure adequate oil supplies. Finally, in mid-May, the transmission was considered ready for the start of the 400-hour durability test.

During this previous testing a large amount of performance data was accumulated and typical results are shown in Figure 37. Correlation of data points with predicted values (solid lines) was outstanding. The need for determining variations from predicted performance and making design modifications did not occur and all effort could be concentrated on the durability phase.

2. Four Hundred Hour Dynamometer Test

On 12 May 1973 durability testing began with transmission SN 10 and was concluded on 28 September 1973. To provide a load schedule consistent with anticipated vehicle requirements, the test schedule (see Appendix C, HMPT-500 Durability Test) was based on measured data procured several years previously. In August 1969, an M113A1 vehicle equipped with a General Electric HMPT-100-2 transmission and a Detroit Diesel 6V53 engine was tested at the Aberdeen Proving Ground Automotive Test Facility. The vehicle was instrumented to determine sprocket load torques and engine power when operated over test courses specified by the MICV Project Manager.

To provide realistic conditions for dynamometer testing, a load schedule exceeding the measured horsepower and torque values was formulated at that time (Figures 38 and 39). These loads were converted to MICV values by direct ratio of the horsepower from 215 to 450 and by assuming a vehicle of

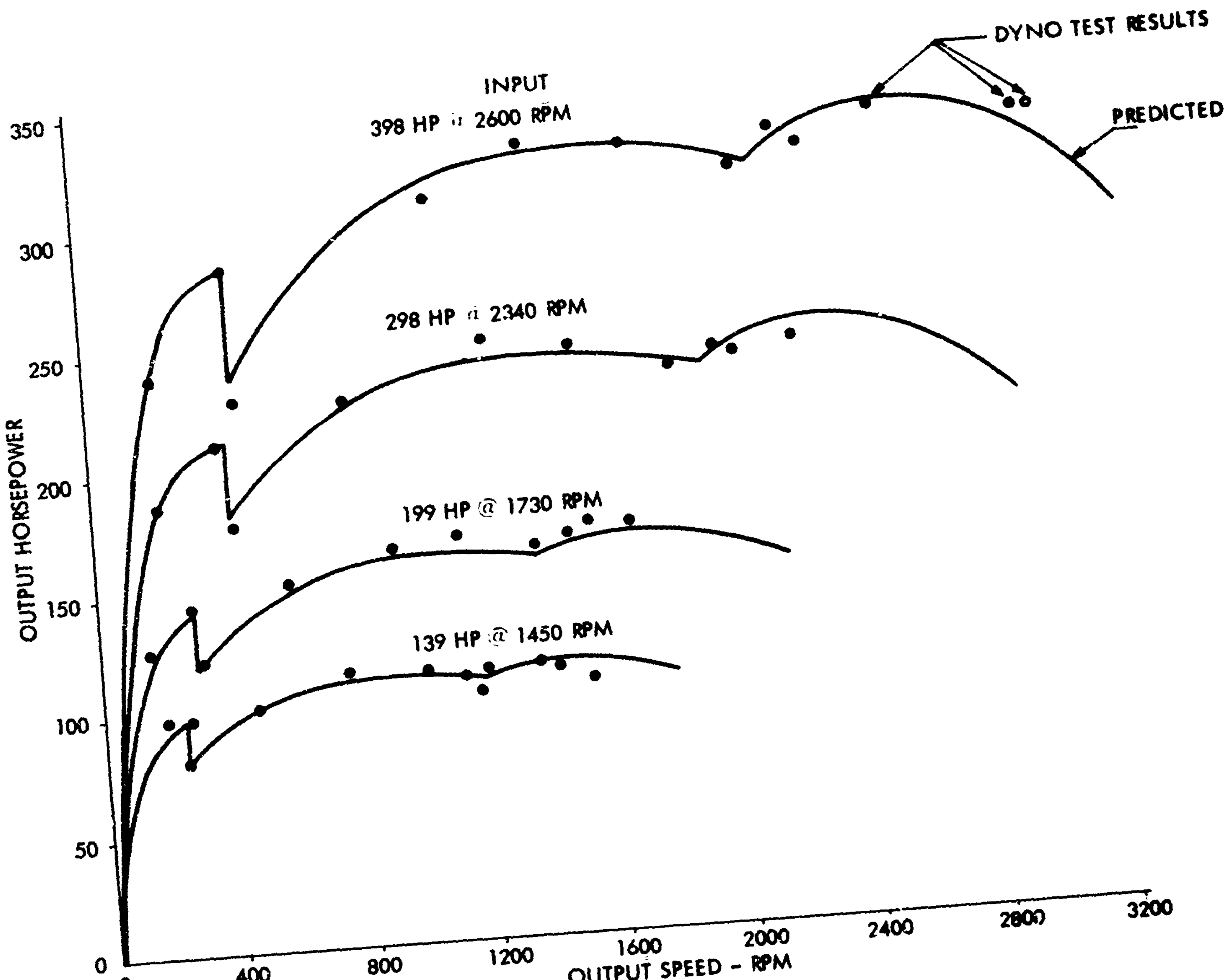


FIGURE 37 HMPT-500 Transmission Performance

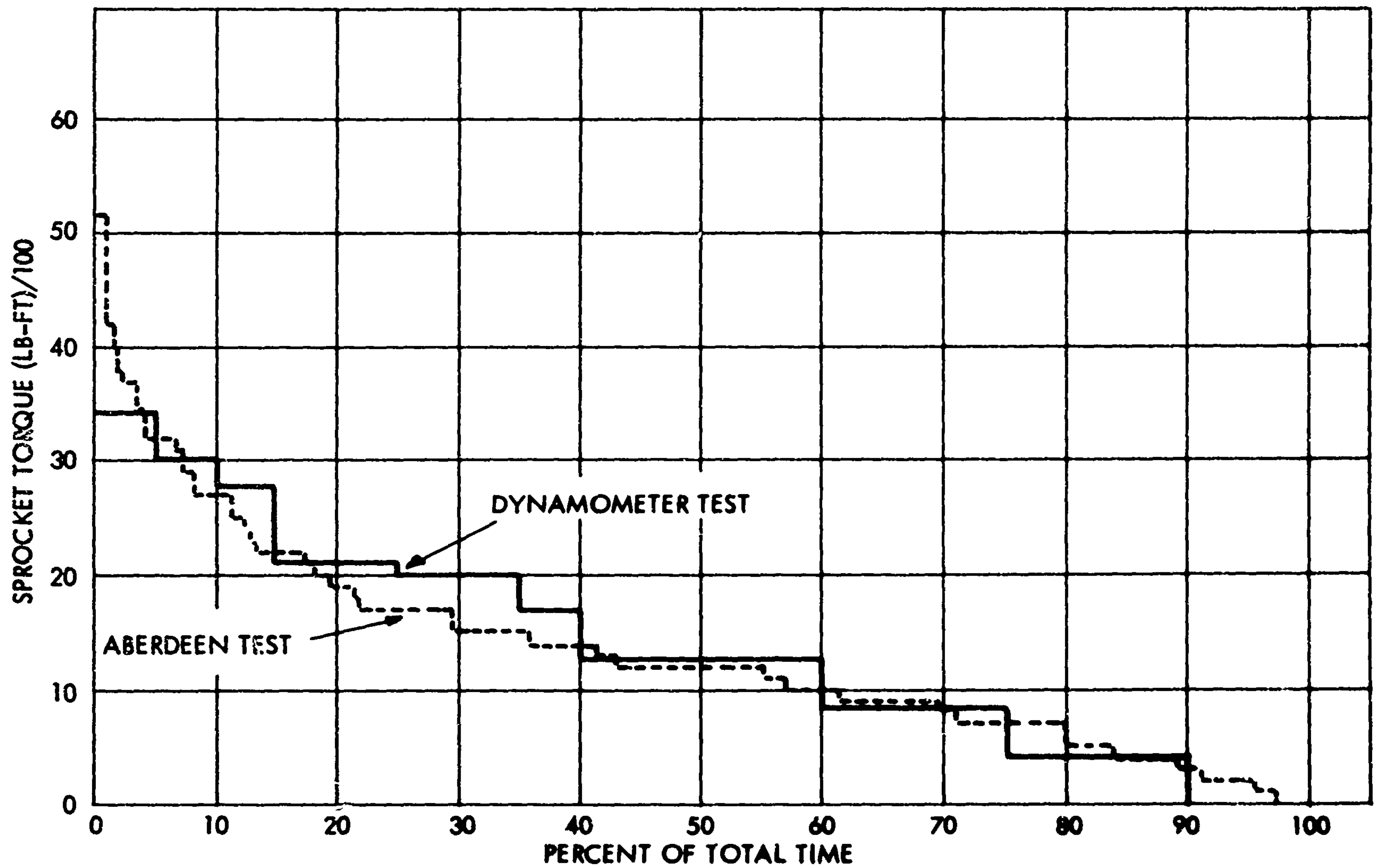


FIGURE 38 Cumulative Torque Distribution per Sprocket

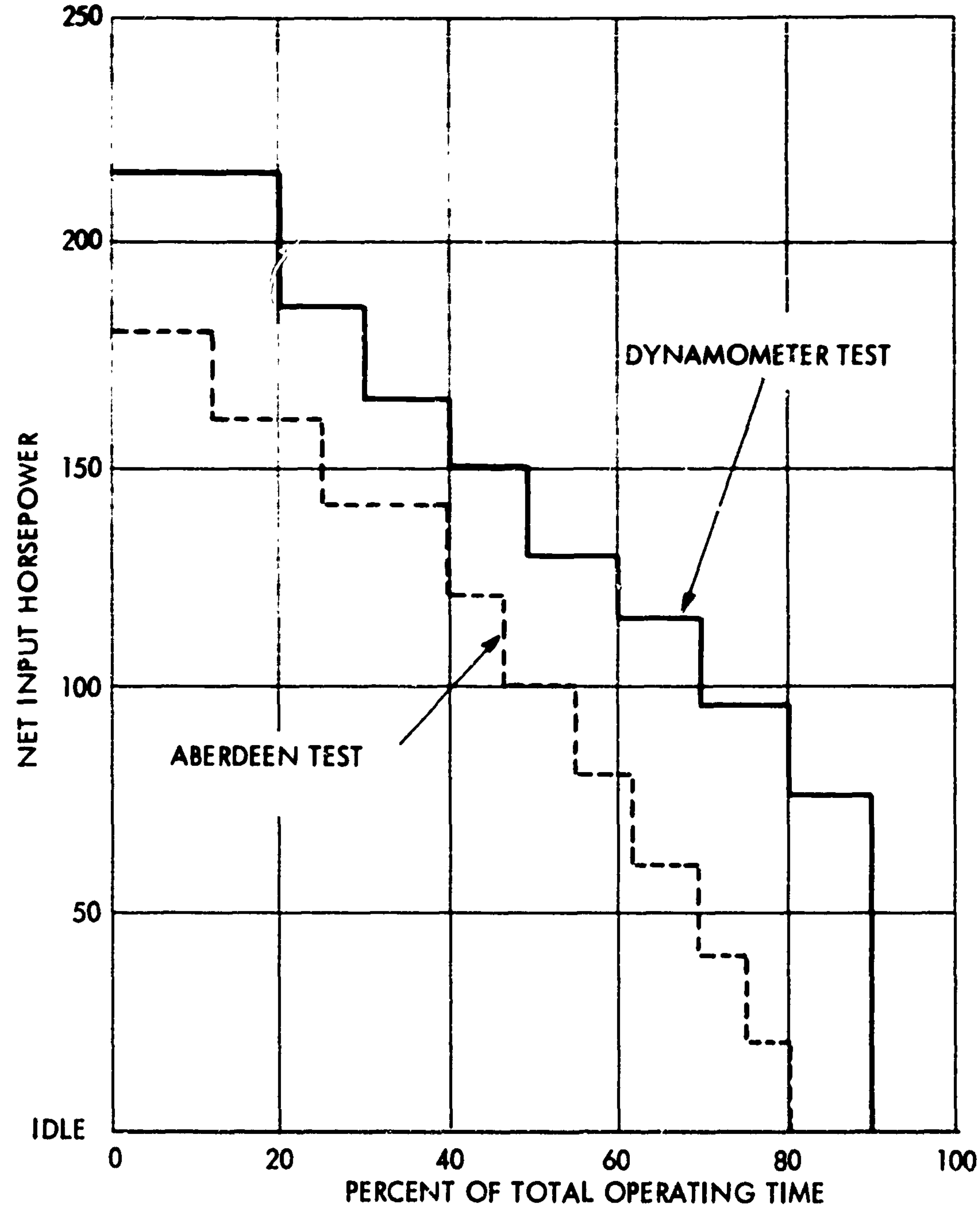


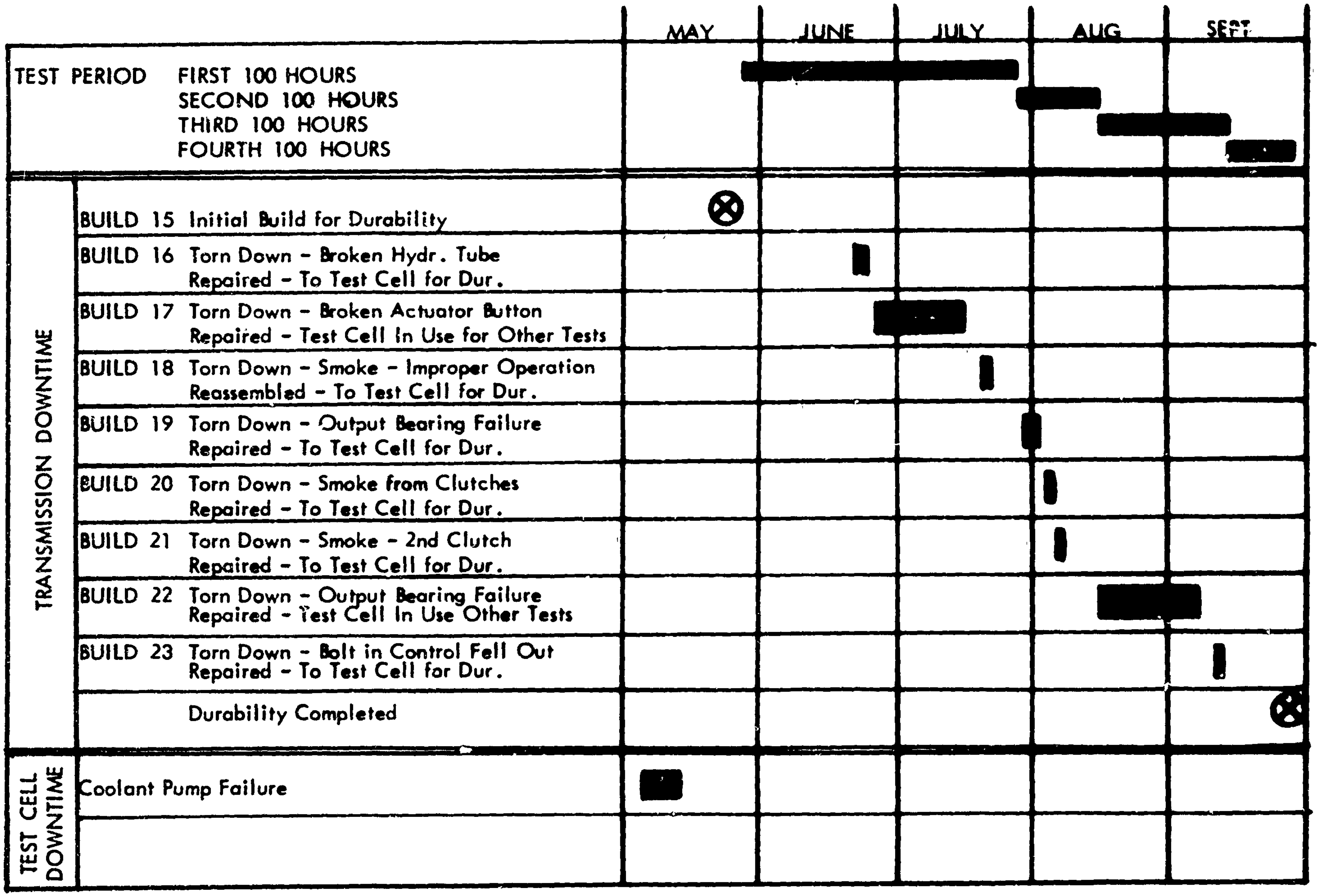
FIGURE 39 Cumulative Power Input

40,000 lbs gross weight, sprocket radius of 0.896 ft, and a final drive ratio of 4.4 to 1 to ratio the torque values. By introducing time at each power and torque point, the load schedule for these tests was broken down into 15 test points totaling five hours for a complete cycle as shown in the Durability Test Plan.

A breakdown of durability test periods and transmission downtime is shown in Table 2. During the total test period, the transmission was torn down eight times for a variety of malfunctions. Brief comments on each teardown are listed below.

- Teardown #16. Broken steel tubing. Material will not withstand vibration. All similar parts replaced with high pressure flexible hose.
- Teardown #17. Broken actuator button and actuator piston seized. Fretting problem on actuator piston. Balance grooves added to piston.
- Teardown #18. Smoke from first and second range brakes. Improper operating procedure in test cell. Operator attempting to maintain a test point at the 1-to-2 shift point.
- Teardown #19. Output bearing failure. Cause obscure at the time. Lubrication holes partly covered by lip seal were opened up.
- Teardown #20. Smoke from brakes. Controller cam improperly machined. Allowed pilots to go from third range to first range at maximum ratio conditions.
- Teardown #21. Smoke from second brake. Assembly tool left in controller which prevented proper brake pilot valve operation.
- Teardown #22. Output bearing failure. A repeat of a previous failure.

TABLE 2 - DURABILITY TEST PROGRAM - SN10



Final conclusions were that bearing cages were defective in batch of bearings from one manufacturer.

- Teardown #23. Bolt in disconnect clutch pilot valve in the control fell out, allowing clutch to disengage while on test point. Proper locking bolt was not available. Bolt safety wired until correct bolt became available.

Of the eight failures, six could be traced to incorrect or defective parts and to human errors. The remaining two, broken tubing and fretting of actuator pistons, were of the type that would normally be associated with long-term durability testing.

Following the 400-hour test, the transmission was completely disassembled for inspection of every component. During disassembly, brake and clutch running clearances, as well as breakaway torques on all bolts, were checked. During this inspection no major discrepancies were detected. Problems found were primarily minor rubs (wear on a thrust washer on one of the A-end blocks, some distress on both second range planet carrier pinion shafts, and slight looseness on the press fit races of two ball bearings). Each of these areas was marked for design modifications on future assemblies. However, based upon the results of dynamometer tests, it was concluded that the transmission design was fundamentally sound and that both efficiency and reliability goals could be met with a minimum of change.

3. Ten Thousand Mile Vehicle Test

The vehicle specified for the 10,000-mile durability test was an XM701 (MICV 65) with a steel hull. Work to modify the vehicle to accept the HMPT-500 and a Detroit Diesel 8V71T began in late 1972. No attempt was made to conform to an "operational" vehicle configuration; rather it was treated strictly as a test bed for power pack testing with ease of installation and removal a prime requirement.

The cooling system utilized a specially-made radiator with a frontal area of about 14 sq ft, combined with the twin fans used in the original vehicle power package. The fans were driven from the crankshaft pulley and a right angle drive at the front of the engine. Transmission oil cooling was accomplished through the use of the oil-to-water cooler which is a normal engine component.

To eliminate major hull changes, intermediate gear boxes (ratio 1:1) were used to adapt the transmission and the vehicle final drives. Transmission saddle mounts were built into these intermediate gear boxes. In the modified configuration the vehicle gross weight with one-half tank of fuel was 40,420 lbs.

When transmission SN 11 was assembled in March 1973, it was sent to the spin stand for check-out, disassembled for inspection, and installed in the vehicle following reassembly. First vehicle operation occurred on 26 April 1973 and, following initial check-out and adjustment, it was driven to the General Electric test track on 2 May 1973. The next five-week period was spent in a cautious evaluation of vehicle power plant characteristics to establish those operating parameters necessary for smooth vehicle operation, primarily clutch pilot valve timing for all operating conditions.

Initially the 8V71T engine had approximately 640 gross horsepower. While this provided rather outstanding vehicle performance, it was felt that the horsepower/weight ratio was not representative of the class of vehicle being simulated. In early June the fuel injectors were replaced so that gross horsepower was reduced to about 550.

Durability testing began in earnest in mid-June and continued until completion on 2 February 1974. During the entire test period the transmission was removed 10 times for repair as indicated by the following:

Teardown	Date	Mileage	
1			Initial Pack Installation
2	5/ 7/73	2.3	First Range Relay Valve Stuck
3	5/25/73	150	Auxiliary Pump Idler Gear Out of Mesh
4	6/21/73	227	Tow Pump Gear Out of Mesh
5	7/ 6/73	319	Second Range Relay Valve Stuck
6	8/ 8/73	1700	Second Range Relay Valve Stuck
7	8/22/73	1850	Second Range Relay Valve Stuck
8	8/29/73	2110	Left-hand Service Brake Plates Warped
9	10/ 3/73	4558	No Failure. Inspection Made of Transmission.
10	12/11/73	8077	Third Range Clutch Plates Burned.
11	1/21/74	9529	Left-hand Output Carrier Internal Spline Worn Out

Each of these primary failures is discussed briefly below.

- Teardown #2. First range relay valve stuck. Bench tests indicated that the relay valve would not release if brake supply pressure exceeded 400 psi. Oil balance grooves were added to the spool and provided proper release at all pressures.
- Teardown #3. Auxiliary pump idler gear came out of mesh after the

retaining ring came out of its groove. Changed to a different type of retaining ring.

- Teardown #4. Tow pump gear came out of mesh. The tow pump idler shaft came loose and backed out of housing allowing the gear to come out of mesh. Both tow pump idler shafts were reinstalled using Loctite and set screws.
- Teardown #5. Second range relay valve stuck due to a metal chip in bore.
- Teardown #6. Second range relay valve stuck due to dirt in the valve. Attempts to continue operation by an inexperienced driver caused burning of the disconnect and third range clutches resulting in further contamination in system.
- Teardown #7. Second range relay valve stuck due to dirt in the valve. The aluminum valve bodies were replaced with a new steel design.
- Teardown #8. Left-hand service brake plates were warped. The vehicle was driven with parking brakes locked which caused the overheating and warping of the brake plates.
- Teardown #10. Third range clutch plates were burned. The spacer on the clutch piston came loose and jammed (not latest design revision). Parts were brought up to the latest revision.
- Teardown #11. Left-hand output carrier internal splines were worn out. This was caused by an alignment problem due to wear between mounting saddle and trunnions. Parts were shimmed to restore proper alignment.

In comparison to dynamometer testing, the failures induced during vehicle tests would probably be considered to be of greater severity. However, this is a normal expectancy based upon the greater variety of transient and shock loads that are induced during vehicle operation.

Upon completion of the 10,000-mile test, the transmission, including controller, was completely disassembled for a critical design evaluation by Transmission Projects Engineering personnel, by consulting engineers from other General Electric departments, and by MICV Project Manager's representatives.

As with the 400-hour durability test, discrepancies found by the reviewing teams were minor in nature and, in most cases, had already been corrected by design changes implemented after the beginning of the 10,000-mile test. Conclusions reached as a result of the test were that the performance of the transmission in the vehicle met all design specifications and that no design defects had been uncovered which would prevent future units from achieving the projected reliability goals.

APPENDIX A

SCOPE OF WORK

CONTRACT DAAE07-72-C-0200

1.0 SCOPE OF WORK

1.1 The contractor, as an independent contractor and not as an agent of the Government, for the period set forth in Section H of this contract, shall furnish the supplies and services necessary to accomplish the engineering and related functions for the development, fabrication, and test of two 500 horsepower ball piston hydromechanical power train (HMPT-500) units for use in the Mechanized Infantry Combat Vehicle, XM 723, MICV program as is hereinafter set forth.

1.2 The design requirements for the HMPT-500 power train are as follows:

a. Net input power capacity of 450 horsepower with a design goal of accommodating 500 horsepower.

b. Maximum input speed adaptable to rated engine speeds of 2400 rpm to 3000 rpm, nominally to be 2600 rpm, by minimum input gear change.

c. The unit, when coupled to a suitable horsepower engine and with typical horsepower/torque data, as illustrated in Figure 1, and with an optimum combination of sprocket pitch diameters of 17" to 23" and a final ratio of 3.0 to 5.0, shall accelerate a 40,000 pound vehicle from 0 to 30 mph in 18 to 22 seconds, reach a top speed of 40 to 45 mph and a reverse speed of 5 to 10 mph.

d. Provide full power forward to reverse shift capability to permit vehicle rocking.

e. Maximum input torque 1100 lb-ft.

f. Maximum sustained total output torque 9300 lb-ft both forward and reverse.

g. Maximum steering torque capability of 5600 lb-ft to either output.

h. Provide integral brakes. Flexibility is desirable with minimum change for either front or rear drive.

i. The mechanical brakes must be capable of holding a 45,000 pound vehicle on a 60 percent slope, and at least one stop from 45 mph at an average rate of 18 ft/sec^2. The unit as a whole must be capable of decelerating at this same rate 25 times from 50 mph at three-minute intervals before brake adjustment. Hydrostatic or hydrodynamic brake assist may be used to meet the sustained number of deceleration requirements.

j. Provide pivot turning about the vehicle center line and steering ability while coasting with the engine inoperative.

k. Provide a power takeoff conveniently located and capable of delivering full horsepower.

l. Provide for push starts as well as towing speeds up to 20 mph for a distance of 25 miles without damage in either forward or reverse.

m. The units must be watertight, corrosion and fungus protected for operation in extreme temperatures of -70° to 125°F, storage from -70° to 155°F, and operate on standard MIL-L-10295 and MIL-L-2104 lubricants as applicable.

n. The unit must operate on 70 percent slopes fore and aft and 40 percent side slopes.

o. The unit shall have, as a design goal, the capability to interface with the following MICV requirements:

1) Engine and Power Train Durability. The vehicle shall have a .50 probability of completing the first 6000 miles of operation without replacement or overhaul of the engine, transmission,

steer system, and final drive. Replacement is considered to have occurred when repair or corrective action exceeds the functions as defined by the approved maintenance allocation chart at organizational and direct support levels.

2) Automotive Subsystem Reliability. The automotive subsystem consists of all components, accessories, and assemblies required to provide an integrated unit capable of satisfying the mobility characteristics of the MICV requirements. The automotive subsystem will have not less than .95 probability of successfully completing a 50-mile mission throughout the first 6000 miles of vehicle operation.

3) Noise. With the vehicle operating in the 10 to 20 mph speed range, the sound pressure levels shall not exceed those shown below at 50 meters. The vehicle shall be operated at all speeds in the above speed range, including climbing a 10° incline and turning to determine highest levels.

Octave Band, Hz	31.5	63	125	250	500	1000	2000	4000	8000
SPL	75	85	82	77	71	68	63	60	55

This represents aural security at 2000 meters with a very quiet background noise.

p. The goal is to keep the total weight under 1300 pounds.

q. "T" configuration and No. 2 SAE BELL HOUSING.

r. Approximate width - 40 inches

Approximate height - 12 inches from output center line

Approximate length - 31 inches

Input to output approximately on center line

s. The distance from the center line of the output shaft to bottom of housing, rear of housing, and radius in between will not exceed 8.5 inches.

t. Provide fully-automatic load sensing controls to select proper ratio to maintain maximum overall efficiency for any load conditions, plus manual override for emergency full power. Control operation will be similar to those of the transmission, cross drive, hydromechanical 250 horsepower, XM1.

1.3 Fabrication. The contractor will fabricate two (2) power train units to the specification outlined in paragraph 1.2 above and the necessary repair parts to support tests outlined in 1.4 below.

1.4 Test. One unit manufactured under this contract is to be subjected by the contractor to a NATO-type dynamometer evaluation for a period of 400 hours. The second unit is to be installed into a tracked vehicle, furnished by the Government, and subjected to a 10,000-mile durability test at the Contractor's test facility.

1.5 Data. Preparation and delivery of the data required and described by Exhibit A.

APPENDIX B

GEAR CALCULATIONS

(from computer program)

SPUR GEAR CALCULATIONS * RUN IDENT MICV - 1st

		Input Data
Number of Teeth, Driver	-	47.0000
Number of Teeth, Driven	-	54.0000
Center Distance	-	8.4166
Pressure Angle	-	19.9988
Diametral Pitch	-	6.0000
Short Addendum - Driver	-	0.
Short Addendum - Driven	-	0.
Minimum Backlash	-	0.0050
Maximum Backlash	-	0.0090
Face Width - Driver	-	1.3750
Face Width - Driven	-	1.3200
Input rpm	-	1.0000
Horsepower	-	1.0000
Input Torque	-	38000.0000
Radius - Cutting Tool Tip	-	0.0400
Load Distribution Factor	-	1.0000
Velocity Factor	-	1.0000
Application Factor	-	1.0000
Material Constant	-	0.0528
Friction Factor	-	0.0600
Plasticity Factor	-	1.0000
Inlet Oil Temperature	-	250.0000
Surface Finish-Micro-In	-	32.0000

Output Data

Gear Ratio	-	1.1489
Whole Depth	-	0.4000
Clearance	-	0.0667
Circular Pitch	-	0.5236
Oper. Diametral Pitch	-	6.0000
Oper. Pressure Angle	-	19.9988
Oper. Circular Pitch	-	0.5236

		Driver	Driven
Oper. Pitch Diameter	-	7.8332713	8.9999287
Pitch Diameter as Cut	-	7.8333333	9.0000000
Addendum	-	0.1667	0.1667
Thickness - Op. Diameter - T Max	-	0.2593	0.2593
Thickness - Op. Diameter - T Min	-	0.2573	0.2573
Thickness - Cut Diameter - T Max	-	0.2593	0.2593
Thickness - Cut Diameter - T Min	-	0.2573	0.2573
Dedendum	-	0.2333	0.2333
Base Diameter	-	7.3609255	8.4572335
Tip Diameter - Maximum	-	8.1666	9.3333
Tip Diameter - Minimum	-	8.1616	9.3283
Tip Diameter - Eff.	-	8.1586	9.3253
Root Diameter - Maximum	-	7.3716	8.5383
Root Diameter - Minimum	-	7.3516	8.5183
Form Diameter	-	7.5800	8.7437
Deg. Roll at - Form Diameter	-	14.0822	15.0398
Deg. Roll at - Pitch Diameter	-	20.8525	20.8525

		Output Data		
		Driver	Driven	
Deg. Roll at Tip, Diameter-Min	- -	27.4410	26.6651	
Deg. Roll at Tip, Diameter-Eff	- -	27.3869	26.6169	
Meas. over Pins - T Max	- -	8.2271	9.3938	Pin Size
Meas. over Pins - T Min	- -	8.2221	9.3938	0.2880
Pointed Tip Diameter - T Max	- -	8.4137	9.5909	
Pointed Tip Diameter - T Min	- -	8.4098	9.5869	
Arc. Thick. at Tip - T Max	- -	0.1283	0.1297	
Arc. Thick. at Tip - T Min	- -	0.1239	0.1253	
Worst Load Diameter	- -	7.8731	9.0438	

Output rpm	-	1.
Output Torque	-	43660.
Tangential Force	-	9702.
Separating Force	-	3531.
W/F =	-	7350.
K =	-	1868.
Hertz Stress Constant	-	246980.
Unit Load	-	44101.
Tooth Mesh Frequency	-	1.
Pitch Line Frequency	-	2.
Contact Ratio - Approach	-	0.8647
Contact Ratio - Recess	-	0.8531
Contact Ratio - Total	-	1.7178
Approach Sliding Velocity	-	0.4259
Approach Specific Sliding	-	0.2077

		Output Data	
Recess Sliding Velocity	-	0.4201	
Recess Specific Sliding	-	0.2049	
Sliding Velocity at	-		
Start of Single Tooth Contact	-	0.0617	
End of Single Tooth Contact	-	0.0559	
Scoring Index	-	1.9222	
Kelly Flash Temperature Data		Delta-T	Final-T
First Point of Contact	-	3.4890	255.7219
Last Point of Contact	-	3.2870	255.3907
First Point of Single Tooth Contact	-	0.4784	250.7846
Last Point of Single Tooth Contact	-	0.4310	250.7069

Lewis Stress Data for Driver

	Tip Load
T	0.3185
H	0.2676
RF	0.0309
Theta-L	25.5915
X	0.0948
Y	0.3791
SB	111664.

APPENDIX C

HMPT-500 DURABILITY TEST PLAN

REQUEST FOR DEVELOPMENT TEST

25 May 1973

I. TITLE: HMPT-500 DURABILITY TEST

II. PURPOSE

To establish transmission component durability characteristics during a 400-hour dynamometer test program.

III. TRANSMISSION TO BE TESTED

HMPT-500 Transmission, SN 10, Assembly 11628220.

IV. TEST SETUP

The transmission is to be installed in the OP 8 Dynamometer Cell with the VT-903 engine and hydraulic connections per 11628320 with the following exceptions or additions:

1. A torque shaft will be installed between the VT-903 engine and transmission for the first portion of test to establish and confirm engine torque output versus speed per control schedule. During the remainder of test, the engine will be coupled directly to the transmission. The engine fuel control will be connected to the transmission and desired engine power versus speed will be provided by a special control cam.
2. The transmission thermostat will be replaced with plug to prevent oil from by-passing the cooler. Temperature of the transmission inlet oil will be controlled by the test stand cooler. Oil temperature to the transmission is to be regulated between 180°F and 200°F for all testing.

V. INSTRUMENTATION

The transmission will be instrumented to provide periodic readout of the following data:

- Engine speed
- Input torque (engine schedule check-out only)
- Output speed (right side and left side)
- Output torque (right side and left side)
- Make-up pressure
- Auxiliary pressure
- Pintle pressure (right and left)
- Oil out (to cooler) pressure
- Oil temperature into transmission
- Oil temperature out of transmission

VI. TEST LIMITS

- Maximum input speed	-	2500 rpm
- Maximum input power	-	450 hp
- Maximum output speed	-	3300 rpm
- Maximum filter drop	-	30 psi
- Maximum sump (oil out) temperature	-	260°F

VII. TEST PROCEDURE

1. The test will be 400 hours, divided into four periods of 100 hours each. At the completion of each 100-hour period, the power train organizational maintenance will be performed. Operator maintenance will be performed each five hours.
2. Oil specified under MIL-L-2104, 30 weight, will be used. An

adequate supply will be available at the commencement of the test and a sample (two 1½ oz. bottles) withdrawn. Thereafter, a similar sample will be taken from the power train every 50 hours. Each sample will be forwarded to the government project engineer for spectrographic inspection.

3. The power train oil filter will be inspected each 50 hours and will be replaced as necessary.

4. Each 100-hour period of the test will be made up of 20 five-hour schedules as follows:

Test Period	Duration minutes	Engine rpm	Input hp (reference)	Output rpm
1	15	1300	150	250
2	20	1100	100	300
3	15	1300	150	1200
4	10	2100	300	350
5	30	1500	200	1200
6	20	1800	250	800
7	20	2100	300	2100
8	5	2430	450	450
9	15	2100	300	1500
10	20	2300	350	2500
11	20	2300	350	1200
12	30	2400	400	1800
13	20	2400	400	2500
14	30	2430	450	1600
15	30	2430	450	3000

5. Output speeds during any test period may vary ± 50 rpm from speed shown for each test period. Input speeds may vary ± 30 rpm during any test period.

6. Data (listed in Section V) will be recorded during the last five minutes of each test period.

7. Except in the case of an emergency, the test will not be shut down

until after completion of any one of the 15 test periods. The test will continue only after a warm-up period sufficient for the oil inlet temperature to reach 170°F. The transmission will be warmed up by operating in first range with output loads not greater than 3000 lb-ft (1500 lb-ft/side). The warm-up period will be recorded, but not credited as test time.